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The Economy Car

ALTHOUGH there has been in the past considerable debate on the subject of both the commercial and technical aspects of the economy car, it is becoming increasingly clear that there is, and will be, a growing world demand for such a vehicle. All the present events point in this direction and Continental constructors have gone quite a considerable distance in the development of these cars. Britain as usual is behind, and relatively little has been done. This situation may not, however, be all to the bad, and although Continental designers have by now a considerable body of practical experience on such designs, much of the general knowledge so gained, is actually available as a guide to automobile engineers in other countries. It should however, now be recognized that the time has arrived when something should be done by the British automobile industry. A truly economical British car should be available in world markets.

A New Outlook

In this connection it would seem that a definite break with the traditional outlook must be made. An entirely fresh approach is, of course, necessary, because the production of a vehicle at competitive prices will not be attained by merely scaling down existing layouts. There is an opportunity to produce something of low cost yet more in line with traditional British quality. While obviously lacking luxury features, the car should nevertheless embody none of the crudities both in detail and appearance that are characteristic of certain of the models produced to date. There would be no need for radio, elaborate separate heater unit, arm rests, steering column gear controls, or even orthodox trim or upholstery. Any surplus cost available should go to refinement and be devoted to such vital work as that of providing silence, and insulation from vibration.

Both from the viewpoint of first cost and servicing, there appears to be a sound case for the grouping of all mechanical components. In other words, front engine, front drive, or rear engine, rear drive, are the obvious starting point. Front drive seems preferable. As the power and transmission unit would be of light weight, this scheme could probably be carried through without the customary penalty of unduly heavy steering. Certain of the Continental vehicles are quite reasonably satisfactory in this connection. One of the advantages of front location is that the luggage capacity is much improved. As compared with the rear engine, the body can, of course, be of better shape and provide greatly increased space.

Although the air cooled flat twin is attractive both commercially and technically, the difficulty here would be the attainment of satisfactory refinement standards. Not only would such an engine be difficult to silence mechanically, but it would probably furnish problems regarding torque reaction. The general effect might be to make the vehicle too rough, particularly by British standards which must aim at providing greater refinement than can be found elsewhere. The conventional two-stroke as a twin or as a three-cylinder would hardly prove sufficiently economical in fuel.

The Bosch petrol injection system recently announced comprising a diesel type pump and cylinder injection as against manifold, may conceivably provide a step forward in two-stroke economy. By eliminating the scavenging waste of the carburetted two-stroke and also improving the firing at light loads, this may have something to offer. Again, however, the important question is cost, and much would depend on this aspect of the project. If a four-stroke engine is employed it would seem that a four-cylinder would be essential.

The Larger Car

As regards larger cars, aiming primarily at economy of operation over extensive mileages, there still seems to be a case for the diesel. It is, however, obvious that the initial outlay must be greater and very considerable mileages would have to be envisaged to make a product of this nature commercially attractive. From the aspect of first cost, the position is still very little changed and there is no sign of any developments that will provide injection equipment at figures sufficiently attractive to bring the vehicle into the range of the moderate mileage of the motorist.

As regards the coachwork, presumably the most suitable layout would be a saloon of chassisless construction. This might be offered with the alternative of the folding roof panel something on the lines of that provided on the Fiat "500". Various economies could be effected in the bodywork detail, such, for example, as windows with simple ratchet type catches giving three or four positions. The rear seat should be readily detachable so that the back of the car could be employed as luggage space for two people on extended travel.

On the subject of suspension, it is possible that with certain of the new developments in mind, an independent layout all round might be devised that would prove to be more economical than axles. It might in fact be essential on a small and ultra-light vehicle, in order to secure satisfactory riding and road holding. The fact that in a chassis of limited wheelbase and track, the independent layout also provides a flat floor, is a considerable added attraction.

To stress once again the essentials of any such project, it would seem that the three outstanding objectives that must be achieved if there is to be any hope of success are, low cost, silence, first-class handling and road holding. In the present state of the art there should be no insuperable difficulty in the attainment of all three. Profiting by experience that is already available and with a knowledge that the Continental industries are going ahead in this field, it seems essential that a start should be made. From the viewpoint of overseas markets the British automobile industry should be in a position to offer a product competitive with those of Continental origin.

British users of economy cars have as a rule preferred a four-door vehicle, but they would doubtless accustom themselves to the two-door type of body, so that it might safely be adopted for economy models. The most serious argument that is at present advanced against the introduction of an economy vehicle relates of course, to material supplies. While existing types of vehicles continue to sell and also consume all the steel that is available, there is a good case for maintaining production on present lines. The material outlook is in fact so unpromising that the introduction of any new model would inevitably involve a reduction in the manufacture of existing types.

It is generally known in the industry that smaller vehicles are in the development stage but that their introduction is indefinitely postponed for a variety of reasons. However, so far as is known, these prototypes are not such as would be likely to be really competitive with the economy cars already being marketed by Continental makers. They correspond much more nearly to scaled down versions of larger cars. In their specifications various desirable features are included that are still regarded as necessary. In this connection, as previously mentioned, the first essential of the true economy car is that such standards must be abandoned. In no other way would it be possible to produce a vehicle competitive with the types made abroad. Nevertheless, the world fuel situation is such that probably the greater part of European motoring in the immediate future years will of necessity be done on vehicles of the economy type.

Taxation

AS was to be anticipated, there has been a sharp, general, and adverse reaction to the budget proposals regarding the motor industry. The Society of Motor Manufacturers and Traders take a very grave view of the new impositions, particularly the disastrous increase in the purchase tax. As evidencing lack of stability and foresight in Government circles, it is regarded as astounding that so soon after the 66⅔ purchase tax on the higher priced cars has been removed, it should be so rapidly reimposed, and on *all* cars.

This latest extortion may well have a calamitous effect upon those manufacturers who were in fact, only saved from disintegration by the abolition of the double tax in the previous budget. The immediate effect on the Home market though serious, is not the whole story. The doubling of the purchase tax must indirectly have an adverse effect on the export market, particularly for the more expensive vehicles. The makers of such cars actually depend on an adequate Home outlet in order that the cost of such vehicles in overseas markets may not be prohibitive.

The additional 4½d. per gallon on fuel is a further blow, but here, the adverse effects are likely to be more widely distributed. In short, such a tax is just a further increase in the cost of living and will affect the whole of the community. Every addition to the cost of road transport is obviously an addition to the cost of production throughout all industry. To the motor industry where raw material supplies are largely dependent on the motor vehicle, this additional burden must mean rises in costs, with the obvious threat to prices and export sales. Those responsible for such policies will find that they have to move with considerable caution in this matter of fuel costs. After all, the Home market though relatively small, is an increasing factor in forward planning. Fuel taxes by focusing attention on fuel consumption may well check the very desirable trend towards the use of larger engines. Higher powered cars are demanded by overseas buyers, and any check on engine sizes must be a retrograde step.

In general terms then, the whole character of the budget in its effect on the motor industry will be towards an increase in prices and will accelerate the inflationary trend. As it is essential in expanding world overseas markets to keep costs down to competitive level, further price increases may well affect dollar earnings. Incidentally, bitter disappointment is felt that the purchase tax on the commercial vehicle chassis has not been abolished.

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THE AUSTIN A.70 AND A.90

Four-Cylinder Chassis of Straightforward Design

BASICALLY very similar in chassis design, the Austin A.70 and A.90 models have established a reputation for performance well above the average for the types of vehicle they represent. So far as the chassis is concerned, the main differences lie in the engine and frame. Although both cars have four cylinder engines, the bore of the A.70 is 79·4 mm., giving a swept volume of 2,199 c.c., whereas the A.90 has a bore of 87·3 mm. giving a swept volume of 2,660 c.c. The A.90 is thus one of the largest four cylinder engines now in production for private cars, but nevertheless it is a smooth running unit providing easy and effortless high speed cruising. All A.90 and A.70 models have an axle ratio of 4·125 : 1.

During the life of the A.70, the coachwork has been entirely re-styled, and a number of chassis modifications have been embodied. Girling hydro-mechanical brakes have been superseded by the Girling full hydraulic system, and on the A.70 the bottom gear ratio has been raised and the wheel base lengthened by 3 in. to 8 ft. 3 in.

Not only is the performance of both models unusually good in terms of acceleration and maximum speed, but also the general handling is in keeping with these potentialities. The gear ratios also are well chosen, allowing satisfactory response when handled by more expert drivers. Constant-load synchromesh mechanism, and not baulking type, is fitted. In this connection it might be suggested, as a general statement, that on cars of the A.90 type the constant-load mechanism is preferable, since when utilizing the maximum performance there are occasions when it is useful to over-ride the synchromesh and "snatch" a change.

SPECIFICATION

ENGINE: A.70—Four cylinders. Bore and stroke 79·4 mm.×111·1 mm. Swept volume 2,199 c.c. Maximum b.h.p. 68 at 3,800 r.p.m. Maximum b.m.e.p. and torque 130 lb. per sq. in. and 116 lb. ft. at 1,700 r.p.m. Compression ratio 6·8 : 1.

A.90. Bore and stroke 87·3 mm.× 111·1 mm. Swept volume 2,660 c.c. Maximum b.h.p. 88 at 3,800 r.p.m. Maximum b.m.e.p. and torque 134 lb. per sq. in. and 144 lb. ft. at 2,000 r.p.m. Compression ratio 7·5 : 1. Three bearing crankshaft. Overhead valves operated by push rods and rockers. A.70, one Zenith carburettor. A.90, twin S.U. carburettors.

CLUTCH: Borg and Beck single dry plate. A.70, 8 in. A.90, 9 in. diameter.

GEARBOX: Four speeds and reverse with constant load synchromesh on top, third and second. Ratios: top, 1 : 1 ; third 1·416 : 1 ; second, 2·25 : 1 ; first, 3·6 : 1 ; reverse, 4·63 : 1.

REAR AXLE: Banjo type with spiral bevel gears. Ratio: 4·125 : 1.

SUSPENSION: Front—independent by wishbones and coil springs. Rear—semi-elliptic springs in conjunction with Hotchkiss drive.

BRAKES: Earlier models—Girling hydro-mechanical with two leading shoes at the front. Later models—Girling full hydraulic with two leading shoes at the front.

TYRES: A.70, 5·50—16 home, 6·00—16 export.

A.90, 5.50—16 Fort.

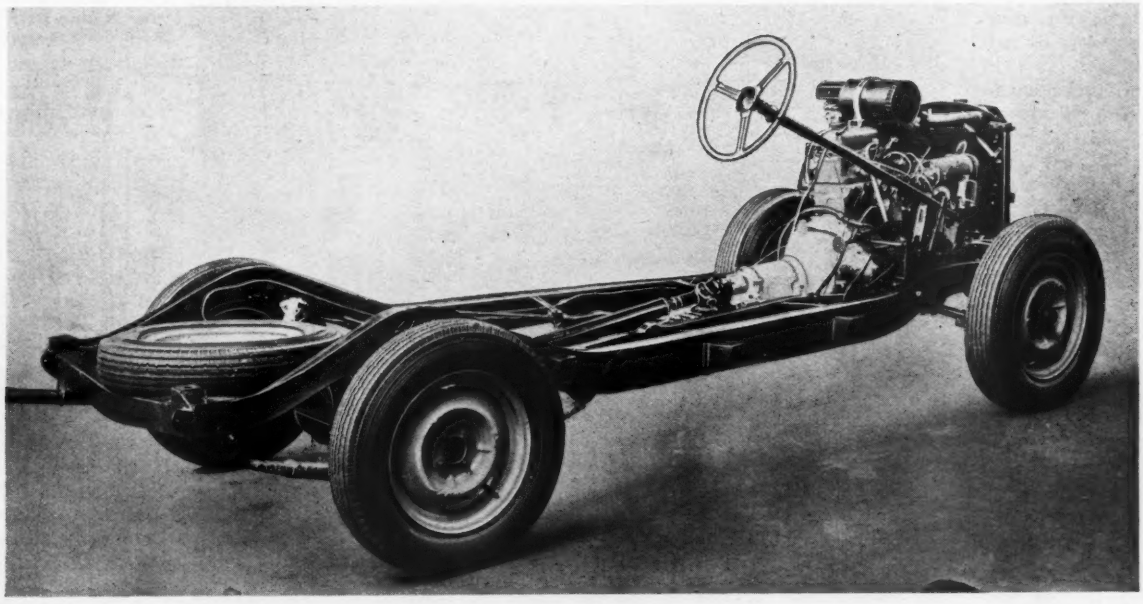
DIMENSIONS: A.70—wheelbase 8 ft. 3 in. Track—front 4 ft. 5 1/16 in., rear 4 ft. 7½ in. A.90—Wheelbase 8 ft. Track—front 4 ft. 5 1/16 in., rear 4 ft. 7½ in.

WEIGHTS: A.70—25 cwt. less petrol. A.90—26 cwt. less petrol.

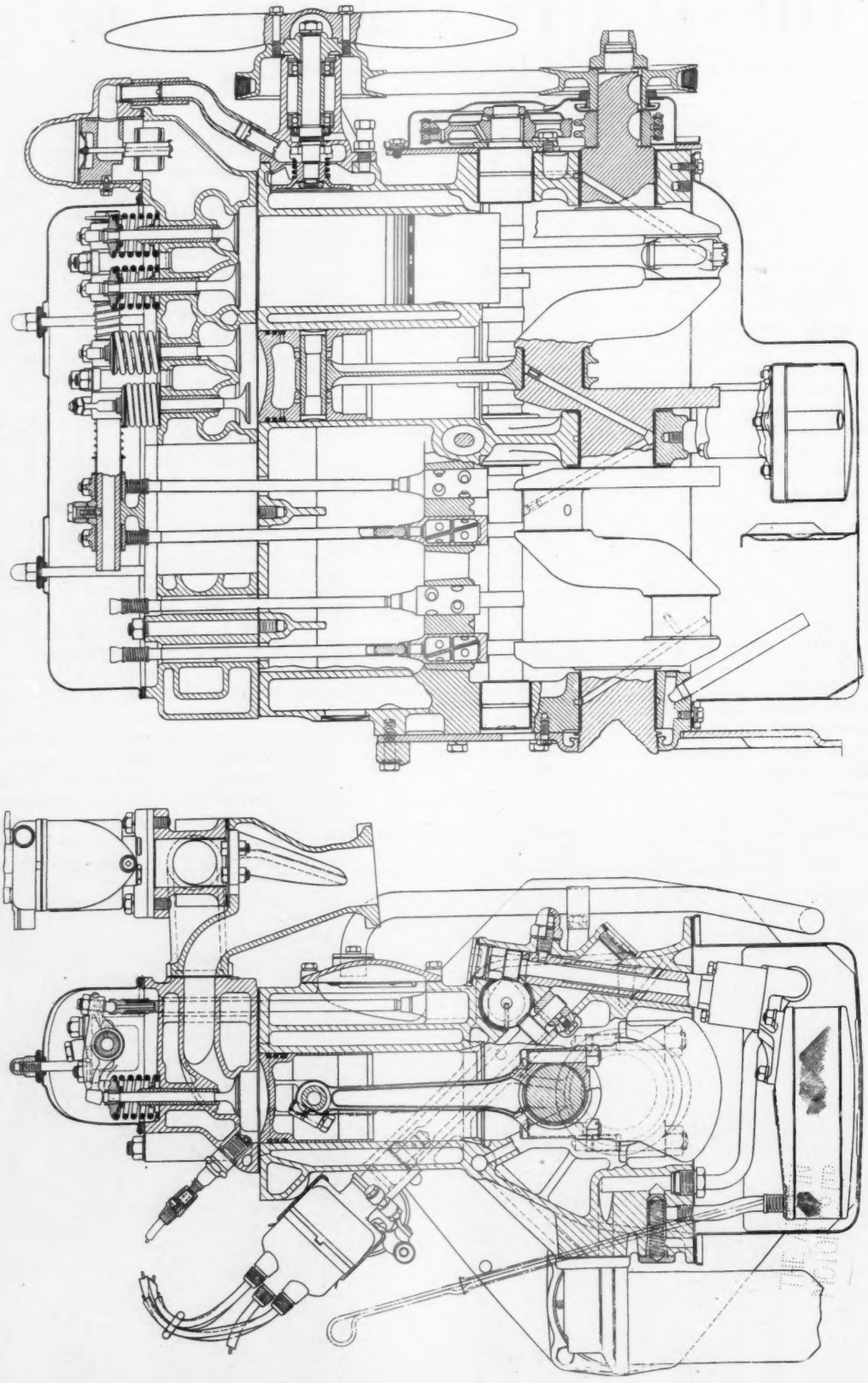
Engine

The four-cylinder engine has a bore and stroke of 79·4 mm. by 111·1 mm. in the case of the A.70, giving a swept volume of 2,199 c.c. and developing 68 b.h.p. at 3,800 r.p.m. with a compression ratio of 6·8 to 1. Maximum b.m.e.p. and torque is 130 lb. sq. in. and 116 lb. ft. at 1,700 r.p.m. The A.90 has a bore and stroke of 87·3 mm. by 111·1 mm., giving a swept volume of 2,660 c.c. and developing 88 b.h.p. at 3,800 r.p.m. with a compression ratio of 7·5 to 1. The maximum b.m.e.p. and torque is 134 lb. sq. in. and 144 lb. ft. at 2,000 r.p.m.

Made in cast iron, the cylinder block and crankcase forms an integral unit extending for 2½ in. below the crankshaft centre line, the three main bearings being well supported by ribbing to the crankcase walls. There are full depth water jackets with water spaces between



The Austin A.70 chassis.



GENERAL ARRANGEMENT OF AUSTIN A.70 ENGINE.
Bore and stroke 79·4 mm. × 111·1 mm.

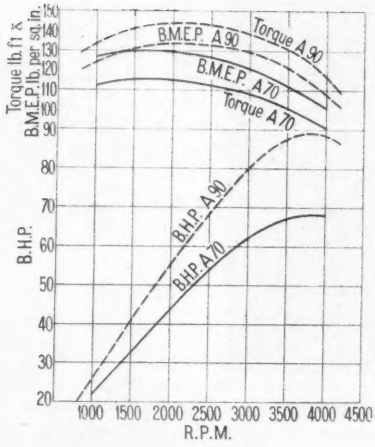
adjacent bores. The cylinder bores are finished by a single point tool, a wire brushing operation being subsequently performed to remove the crests from the resultant slight ridges.

The main bearing caps are retained by studs and the ends of the cap also locate in recesses in the cylinder block. The main bearings are of the steel-back, white metal, thin shell type, each bearing half having two feed holes supplied from circumferential grooves in the bearing housings and caps. The main oil gallery and feeds to the main bearings are formed by drillings in the casting.

Of En 16, the crankshaft has journal diameters of 2·479 in. and crankpin diameters of 2 in. Balance weights are formed integrally on the end and centre webs and end thrust is taken by strip facings on either side of the centre main bearing.

Behind the rear main bearing is the usual oil thrower and return thread, the crankshaft terminating in a flange to which the flywheel is attached. Beyond the front main bearing the forward extension of the shaft carries the camshaft drive sprocket and the fan belt pulley, each located by Woodruff keys and having an oil thrower between them. There are the usual diagonal oil drillings, but in the case of the crankpins there are additional drillings breaking into the diagonal drillings from a point about 120 deg from the main break-through on the journal surface, in the trailing direction. Restricters are fitted in the diagonal drillings so that more oil is induced to flow through the trailing passages which break out in a region of relatively low pressure in the bearing.

Made in En.16, the connecting rods have a centre distance of 8·1875 in. with double ribbed caps located by fitted bolts. Steel-backed, white metal bearing shells are fitted and the gudgeon pin is clamped in the small end. The shank of the rod is offset slightly from the big-end. Split



A.70 and A.90 performance curves.

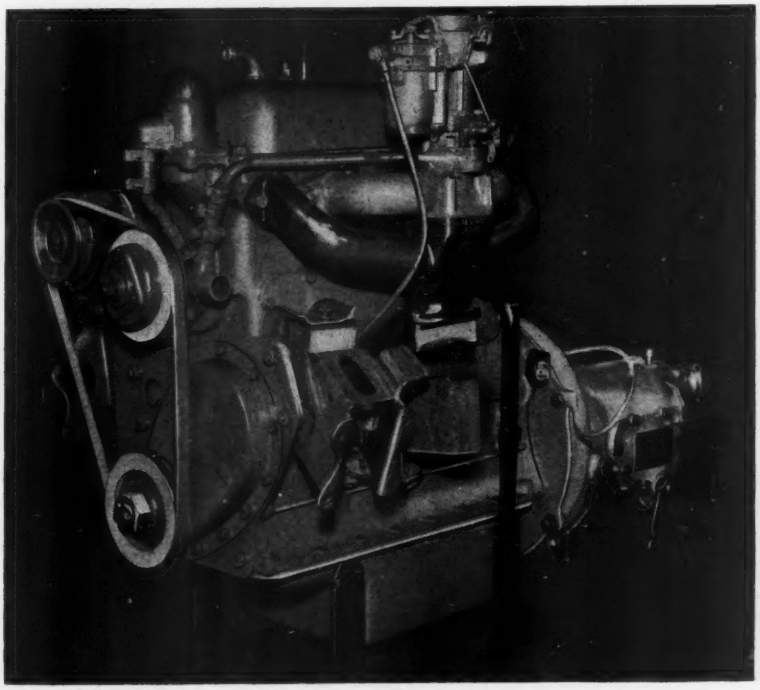
skirt pistons of alumilited aluminium alloy carry two compression rings and one scraper ring. The groove widths are 0·127 to 0·128 in. for the compression rings and 0·190 in. for the scraper rings. The gudgeon pin is a push fit at 70 deg F. Ribs from the gudgeon pin bosses support the piston crown which is slightly dished.

Secured to the cylinder block by eleven $\frac{7}{16}$ in. studs is the cast iron cylinder head. The shape of the combustion chambers is shown in the accompanying illustration and the design was carried out in collaboration with Weslake and Co. Ltd. The inlet port is tapered to increase the velocity of the charge from the induction pipe, to straighten it out and to produce a piled effect by making use of the kinetic energy of the charge. Around the valve guide the port is shaped to decrease the obstruction and at the valve seat there is a radiused entry leading to the valve seat proper. This, in conjunction with the valve shape, gives the highest possible flow at low lifts and the effectiveness of the scheme is very evident in the performance of the vehicle. The exhaust valve has a substantial valve guide boss which, by decreasing the eddying behind the valve head, has been found to increase the flow through the exhaust port. It also permits better cooling of the valve guide and exposes less of the valve stem to the exhaust gases.

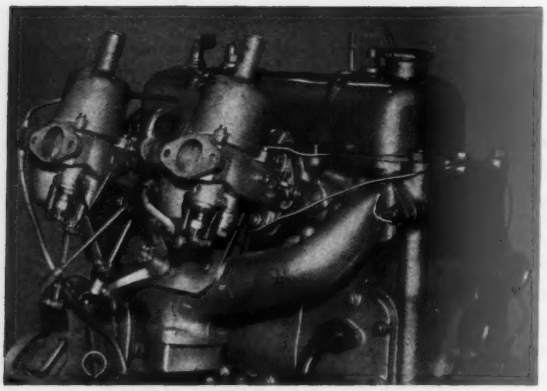
It will be seen that there is considerable freedom in the combustion chamber ahead of the inlet valves. It is stated that the maximum discharge point on most inlet ports is the leading edge, and the back wall of the combustion chamber behind the inlet port can be brought up quite closely with little loss of flow. Between the inlet and exhaust ports projects a peak. This diverts the inlet charge away from the hot exhaust valve and promotes a high degree of rotational swirl.

The sparking plug is placed at the apex of the heart shaped combustion chamber so that the high rotational swirl centrifuges the fuel in a heavy layer around the sparking plug. By this means it is claimed that some degree of stratification of the charge can be obtained. By a combination of the peak and shape of the wall at the back of the inlet valve, various degrees of turbulence can be obtained. On the A.90 the wall behind the inlet valve is curved to increase the air flow and turbulence and to project the fuel forward in the charge and thereby increase stratification.

A gallery system directs the cooling water



Near-side of A.70 power unit.



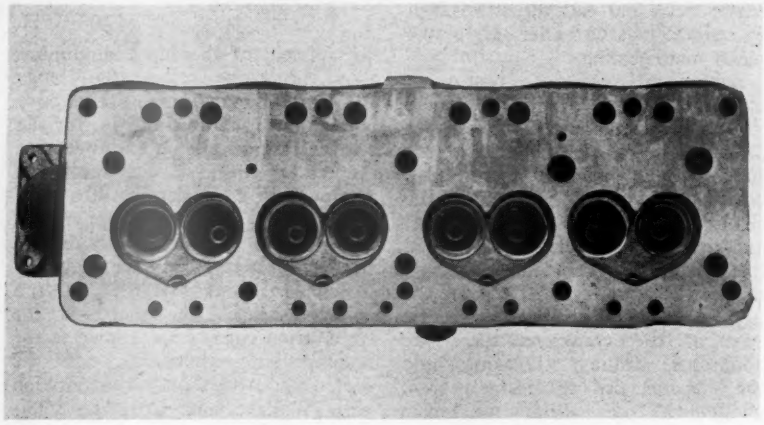
A.90 induction system with twin S.U. carburettors.

around the face of the combustion chamber that carries the sparking plug. After the exhaust valve, this is the hottest spot in the combustion chamber and since the incoming inlet charge must impinge on this face it is essential that it should be fully cooled. There are cross bleeds in the gallery system that feed water into the cylinder block and which are directed across the top of the block.

Mounted vertically in the cylinder head and offset slightly from the centre line of the bores, the valves are operated through rockers, push rods and barrel-type tappets from the three bearing En.32 camshaft carried low down in the near-side of the crankcase. It is driven by duplex roller chain and the camshaft sprocket embodies a synthetic rubber tension ring lying in a recess between the two rows of teeth.

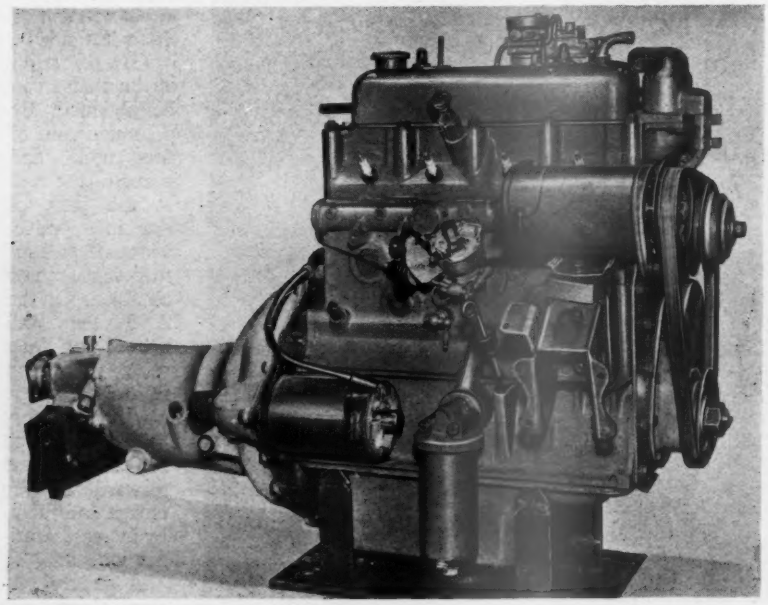
Silicon chrome steel is used for the inlet valves and the exhaust valves are of XB steel. The seat angle is 45 deg and the exhaust valve guides are counter-bored at the combustion chamber end. Single valve springs are fitted, retained by split cotters which are themselves retained on the valve stem, when spring pressure is released, by means of hairpin springs. Valve clearance is adjusted by means of the customary ball-ended screw and locknut, but an unusual refinement is provided at the upper end of the push rod in the form of a cushioning device. The cup end of the push rod is free to slide in the bore of the rod and is provided with a light spring to take up clearance. When under compression, the cup end seats on the blind end of the bore in the push rod, but since the cup is drilled axially a cushion of oil is formed so that while the valve mechanism is in no sense self-adjusting, a considerable degree of silencing is obtained.

The valve timing is as follows: Inlet opens 5 deg before top dead centre.



Cylinder head showing combustion chambers.

Inlet closes 45 deg after bottom dead centre.
Exhaust opens 40 deg before bottom dead centre.
Exhaust closes 10 deg after top dead centre.



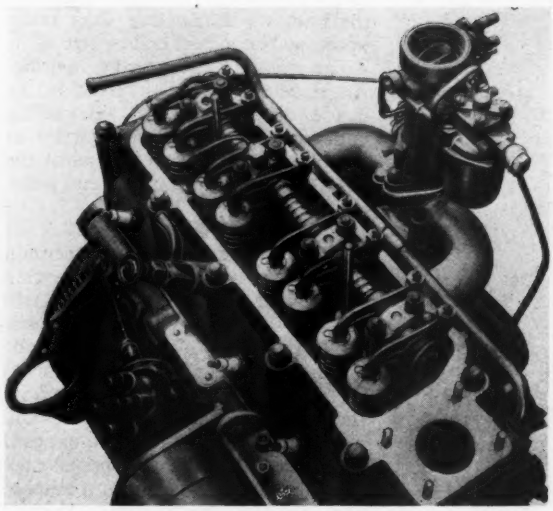
Off-side of A.70 power unit.

Auxiliary drives

A skew gear midway along the camshaft drives the gear-type oil pump. From the top end of the oil pump drive spindle a drive is taken for the revolution indicator fitted on the A.90. Driven from the same camshaft gear is a spindle running diagonally through the cylinder block and terminating in the distributor drive on the off-side of the engine.

Induction system

In the near-side face of the cylinder head are two siamesed inlet ports and three exhaust ports. On the A.70 the inlet and exhaust manifolds are separate castings held together by four set pins below the carburettor flange where a hot-spot is located. A Zenith 42 V.I.S. downdraught carburettor with accelerator pump and economy device is fitted. Twin S.U.



A.70 valve mechanism and induction system.

carburettors are used on the A.90, each carburettor having its own manifold communicating with siamesed ports. A special Austin patented device for use with S.U. carburettors is embodied, to obtain a fast idle speed without an unduly rich mixture. The choke control levers are so designed that as the needle seat is lowered to allow more petrol to flow through, the throttle is opened slightly. There is the usual balance pipe between the carburettors, but no hot-spot is provided. A commendable feature of both cars is the use of a linkage having spring-loaded ball joints for operating the throttle, and not a flexible cable.

Ignition

Coil ignition is employed, with a Lucas distributor having both centrifugal and suction advance mechanism.

Cooling System

Engine cooling is by means of a pump and the customary by-pass thermostat system. In the case of the A.70 the thermostat opens at 65 to 70 deg C and on the A.90 at 70 to 75 deg C. Whereas a by-pass is provided on the A.70, on the A.90 the heater pipes serve this purpose. The radiator embodies the usual Austin expansion chamber.

Lubrication

The oil pump is provided with a suction strainer and delivers through a full-flow Tecalemit or Purolator Micronic filter to the main gallery along the off-side of the engine. A plunger-type relief valve between the pump and filter leads back to the strainer body. From the main gallery, drillings lead across and downwards to circumferential grooves in the housings and caps behind the main bearing shells. From these grooves further drillings lead to the camshaft bearings. Oil is led to the big-ends through the diagonal drillings in the crankshaft described earlier and providing a double feed from the crankpins.

The shoulders of the connecting rods are drilled to provide an oil jet for the cylinder bores. Cross drillings in the front camshaft journal lead oil via longitudinal holes to the thrust plate and sprocket, while from the rear camshaft journal cross drillings provide an intermittent feed to drillings through the block and cylinder head that feed the pipe leading to the rear intermediate rocker bracket. An axial drilling from the rear end of the camshaft leads oil to the centre journal where radial drillings feed similar passages leading to the front intermediate rocker bracket. The rockers are drilled to provide a feed to the adjusting ball. In normal running conditions the oil is maintained at a pressure of 50 to 55 lb. sq. in.

Clutch

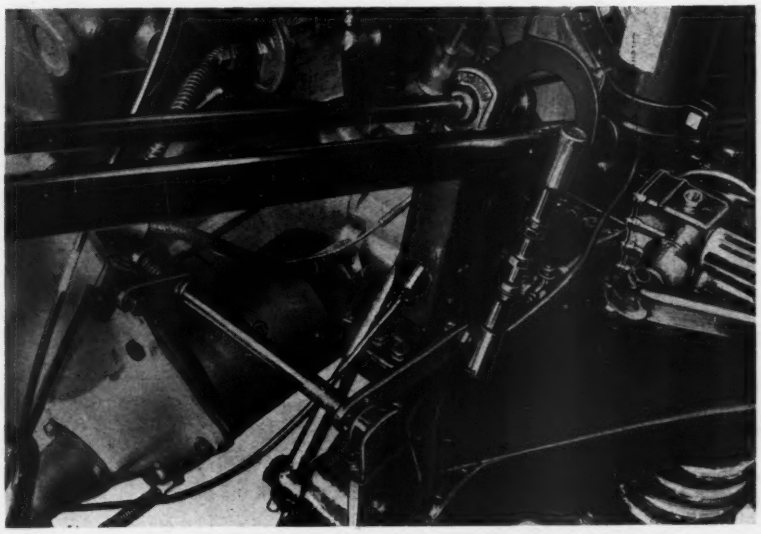
A Borg and Beck single dry plate clutch is fitted, 8 in. diameter on the A.70 and 9 in. diameter on the A.90, giving total friction areas of 45 sq. in. and 65·8 sq. in. respectively. The thrust bearing is of the carbon block type.

Gearbox

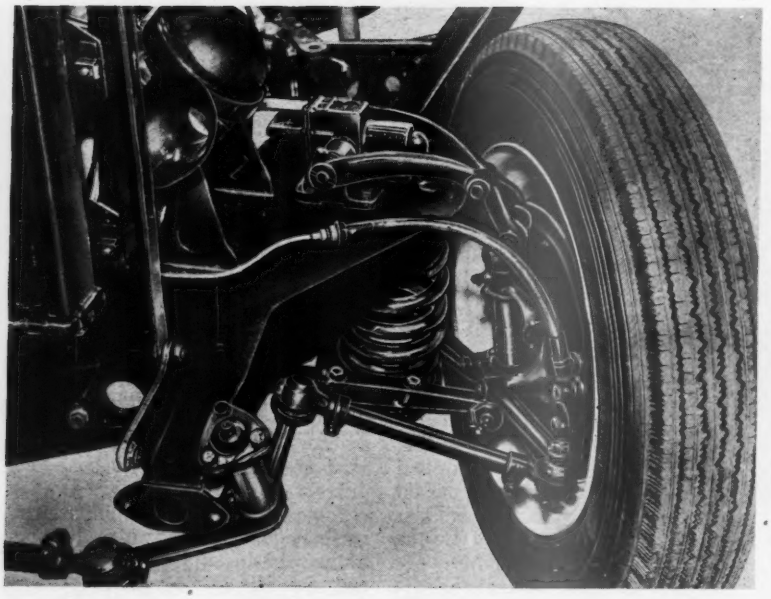
A four-speed and reverse gearbox provides the following ratios : top 1 : 1 ; third, 1·416 : 1 ; second, 2·25 : 1 ; first, 3·6 : 1 ; reverse, 4·63 : 1. All gears except first speed are helical and are made in En.34. Constant load synchromesh mechanism of the spring-loaded ball type is provided on second, third and top, the synchronizer and synchronizer sleeves being made in En.24. The cones have an included angle of 10 deg 20 min and are provided with a coarse thread having a lead of 0·15 in. There are also axial grooves. The major cone diameter is 2·5 in.

A ball bearing supports the primary shaft which is provided with a phosphor bronze bush supporting the front end of the En.34 mainshaft. A second ball bearing supports the mainshaft in the rear wall of the gearbox and a third ball bearing provides an outrigger support in the gearbox rear extension housing. The layshaft gears are formed integrally with a sleeve carried on two Clevite bushes running on the fixed layshaft.

A feature of the gearbox is the oil pump in the form of a threaded sleeve mounted behind the speedometer gear on the mainshaft. Oil thrown off the gears is collected in a trough and fed to an annular space in the rear housing from which it feeds into the thread and is carried forward through a transfer hole into an axial drilling in the mainshaft. Cross drillings deliver oil to the mainshaft gear bearings and the stem wheel spigot bearing.



Steering column gear control linkage.



Steering linkage and front suspension.

The striking forks slide on the selector rods and embody the usual ball locks. A caliper type of selector mechanism is mounted horizontally on the near-side of the box and is operated by a flexible cable from the steering column control. The actual shift mechanism is operated by rods, cranks and ball jointed connectors. The gearbox casing is integral with the bell housing and is made in aluminium alloy.

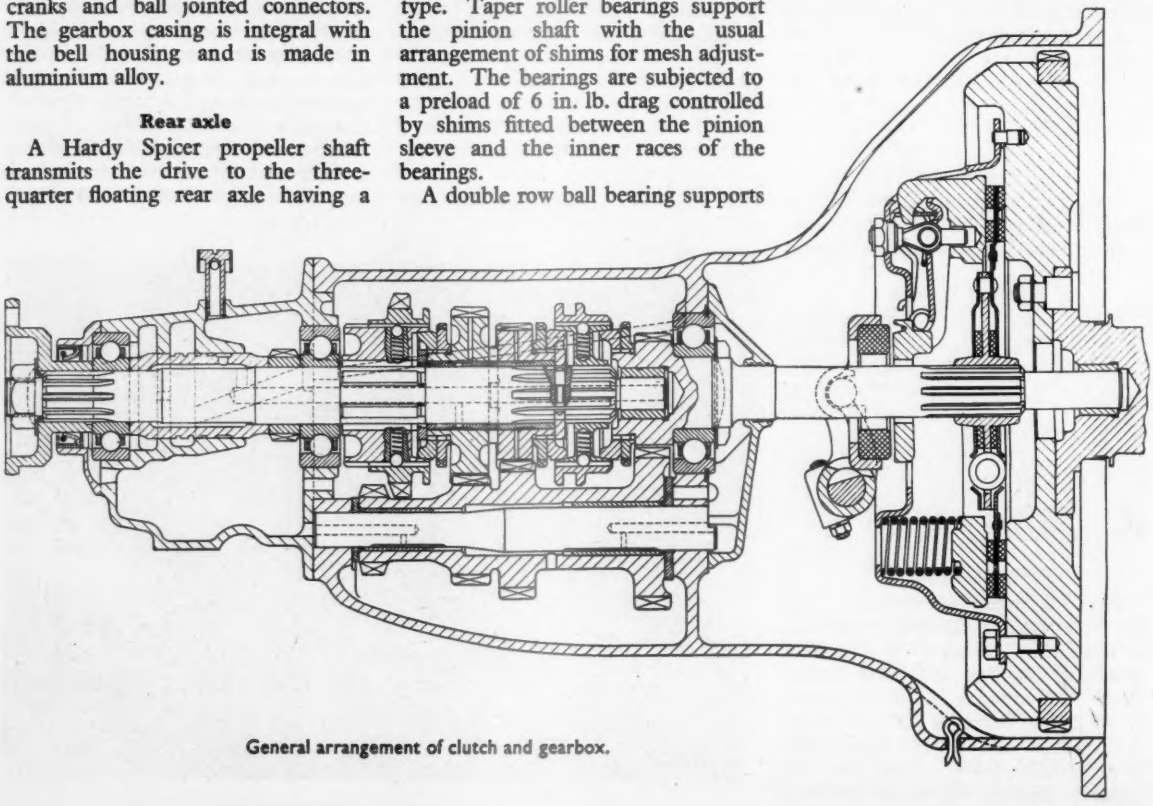
Rear axle

A Hardy Spicer propeller shaft transmits the drive to the three-quarter floating rear axle having a ratio of 4·125 : 1 (8/33). The spiral bevel crown wheel and pinion are made in En.34 and the two-pin differential assembly is carried in a one-piece malleable iron housing bolted to the crown wheel and supported in ball bearings in the axle centre-piece. The axle casing is of banjo type. Taper roller bearings support the pinion shaft with the usual arrangement of shims for mesh adjustment. The bearings are subjected to a preload of 6 in. lb. drag controlled by shims fitted between the pinion sleeve and the inner races of the bearings.

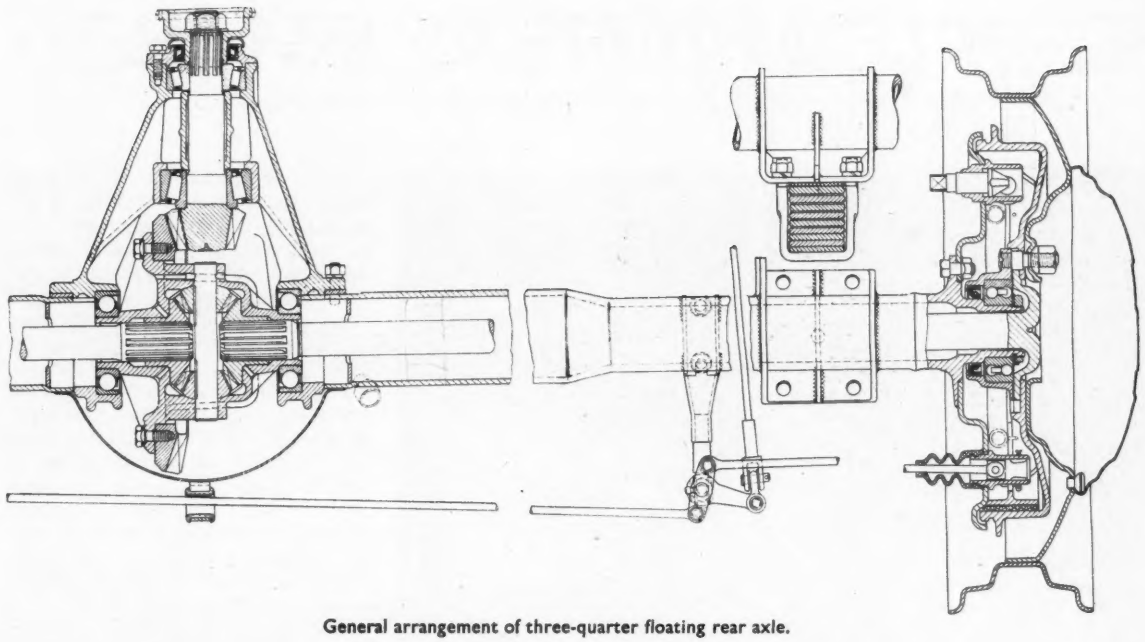
A double row ball bearing supports the hub, the En.16 axle shaft being upset at the outer end where it is flange fitted to the bearing housing. An oil return thread is turned in the inside diameter of the axle casing end piece and an oil seal carried in the bearing housing, seals against the outer diameter of the casing end piece.

Suspension

The independent front suspension is of the wishbone and coil spring type similar to that of the Austin A.40 described in the *Automobile Engineer* for April, 1948. The upper wishbone is formed from two forgings joined together by bolting, and pivots from the damper spindle. At the outer end a trunnion link articulates on opposed tapered and flanged rubber bushes and is retained by a nut on the upper end of the swivel pin. Formed integral with the swivel pin is the lower trunnion on which the lower wishbone forgings articulate on screwed bushes. The lower forgings are connected by a pressed plate forming the spring seating and carrying the rubber bumper. At its inner end the wishbone is carried on flanged and tapered rubber bushings. A castor angle of 1¼ deg is provided and the camber angle is 1 deg. The swivel pin inclination is 6½ deg. A hollow-backed stamping forms the stub axle and



General arrangement of clutch and gearbox.



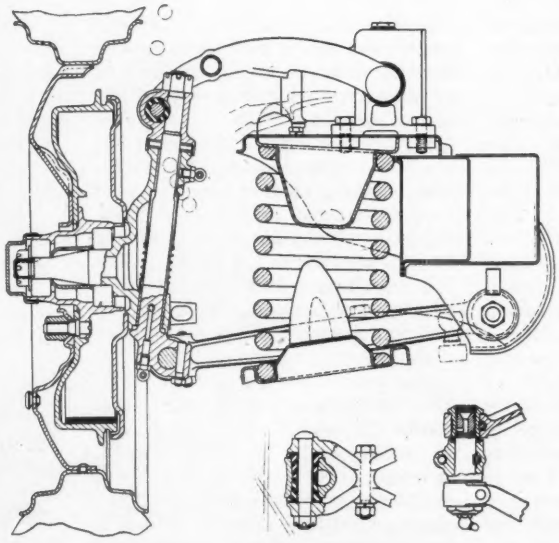
General arrangement of three-quarter floating rear axle.

embodies the brake back plate attachment flange. The swivel pin bushes are widely spaced and are lubricated by separate oil nipples top and bottom. An Oilite washer between two stainless steel washers forms the thrust bearing. Double purpose ball bearings carry the hub.

Having a free length of 11·13–11·43 in. and a fitted length of 7·78–7·98 in., the suspension spring has 8½ coils of 0·621 in. diameter wire. The inside diameter of the coil is 3·599 in. Armstrong hydraulic double acting dampers are fitted.

The Burman cam and lever high efficiency steering box carries a double arm on the drop arm spindle. The rearwardly extending arm is connected by a ball-jointed tie rod to the steering arm attached to the stub. From the forward projecting arm a ball-jointed cross tube connects to the corresponding arm of a slave drop arm mounted on the opposite side of the frame. The rearward extending arm of the slave drop arm is similarly connected to the adjacent stub axle. The spindle of the slave drop arm is carried in phosphor bronze bushes in a housing that also forms an oil reservoir.

Laminated springs are used for the rear suspension and have nine leaves with three zinc inter-leaves. The static deflection is 5½ in., at 784 lb. and the free length between spring eyes is 43 in. Rubber bushes are fitted at the spring eyes but a commendable feature in the interests of lateral stability is the use of phosphor bronze shackle bushes. Armstrong lever-type dampers are fitted, an anti-roll bar being attached by bolts and clips to the damper arms. The springs are protected against corrosion by a binding of gum impregnated tape.



Arrangement of independent front suspension.

Frame

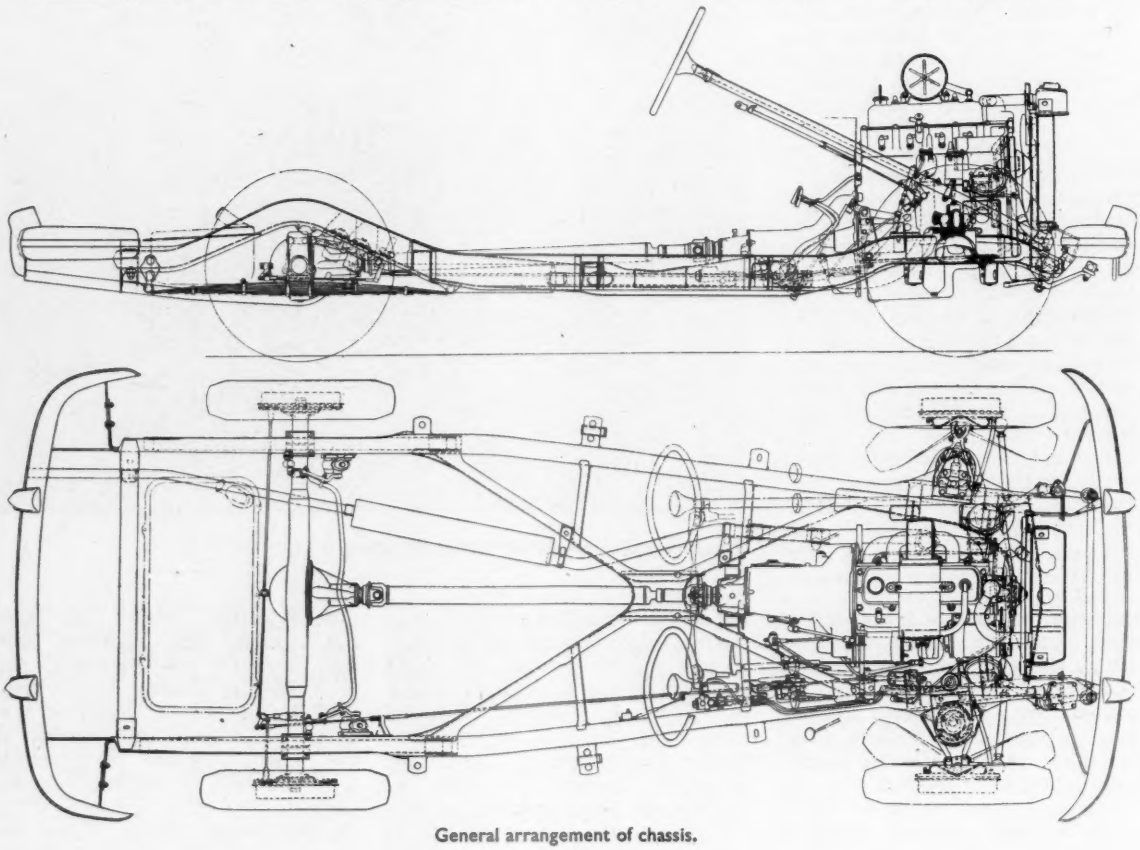
Made in general from 16 s.w.g. material, the frame has large box section side members and front and rear cross members. An X bracing in channel section with the flanges turned outwards has cover plates top and bottom at the junction, where the channels form the sides of the box and run parallel over this distance. The propeller shaft passes above the junction. The forward ends of the side members are stiffened by plates to which the dampers are attached and which form the spring seatings on the under-side. The front engine mounting brackets are also welded to these plates. The mountings themselves are of the rubber-in-shear type with compression block snubbers. Rubber in shear is also used for the rear mounting under the gearbox extension and attached to the X bracing just forward of the junction. Fore and aft movement is controlled by a tie from the frame cross bracket with two rubber blocks in compression. An adjustment is provided on the tie.

Brakes

On earlier models the Girling hydro-mechanical system with two leading shoes was embodied. This has now been superseded by the Girling full hydraulic system. On the A.70 the

brakes are 11 in.×1¾ in., and on the A.90, 11 in.×2¼ in.

Wheelbase, A.70, 8 ft. 3 in.; A.90, 8 ft. Track (front) A.70, 4 ft. 5 ⅛in.; A.90, 4 ft. 5 1/16 in.; (rear) A.70, 4 ft. 7½ in.; A.90, 4 ft. 7½ in. Ground clearance 7½ in. Turning circle, A.70 40 ft., A.90 39 ft. Weight, A.70, 25 cwt., A.90, 26 cwt. less petrol.



General arrangement of chassis.

PRESS GUARDS

SOME time ago, Radiovisor Parent Ltd., 1 Stanhope St., London, N.W.1, introduced a photo-electric press guard incorporating novel features in design. Briefly, the system consists in projecting a "curtain" of light to form a barrier across the opening of the press. The light falls on a "receiver" at the side of the machine and is concentrated on a number of photo-electric cells, and the receiver is connected to a control unit incorporating relays that remain energized as long as the light ray is uninterrupted. Should the operator interrupt the curtain of light by bringing his hand or any part of his body into the danger area, the relays are instantaneously de-energized. This immediately stops the press and makes further operation impossible even though the machine controls are operated.

Such a system, necessarily, protects only the front of the press. In most applications this is perfectly adequate since the ends can usually be enclosed by ordinary fixed guards without any loss of efficiency. There are other applications in which it is essential to guard two or more sides. For example, hydraulic presses generally need to be protected on all four sides. Recently, a hydraulic press measuring 10 ft. 8 in. long and 4 ft. 3 in. deep had to be guarded. An ingenious scheme was adopted to avoid the necessity for employing four separate units, that is, one for each side.

The front of the machine is guarded by a unit situated at the right-hand corner. It projects a barrier of light 16 in. deep and 2 in. wide across the opening. The depth of this barrier is sufficient to provide an ample safety margin when the press is opened to its maximum position. Light from the unit at the front of the press falls on to a 16½ in. × 4¼ in. mirror pedestal-mounted in a pressed steel housing at the front left-hand corner of the press. To permit alignment with the rays, the housing is adjustable. The mirror is fixed at 45 deg to the direction of the beam and, consequently, the light is reflected along the side of the press and falls on to a receiver mounted at the rear left-hand corner.

A second unit projects a light barrier along the opposite side of the machine on to another mirror also set at 45 deg to reflect the beam across the rear to a second receiver. By this arrangement all four sides are fully protected by two units. Interruption of any of the beams causes the press mechanism to stop instantly. This makes the machine safe even if any of the operators are hidden from the man at the controls. (1927)

ACCESSORY MANUFACTURE

The Production Methods at the Works of Wilmot Breeden Ltd.

WILMOT BREEDEN LTD. produce a very wide range of products for the automobile industry, and each class of product includes at least several, and often many, different designs. In itself the variety of products necessitates an organization of considerable flexibility. When, in addition, there are very wide variations in the numbers required, flexibility in organization becomes a matter of major importance to allow the most effective and economic use to be made of the production facilities.

So far as the automobile industry is concerned the main products of Wilmot Breeden Ltd. are bumpers, locks, window regulators, handles, steering wheels and interior fittings. There is also a substantial output of control knobs, lever ends, ash trays, motifs and badges in decorative plastic materials. Because of the need for flexibility, the factory has been organized for process grouping for all products other than bumpers for which product grouping is employed. In fact, a new factory is now in course of construction and will eventually be used solely for bumpers required in large quantities.

Broadly speaking the factory is organized into the following divisions :

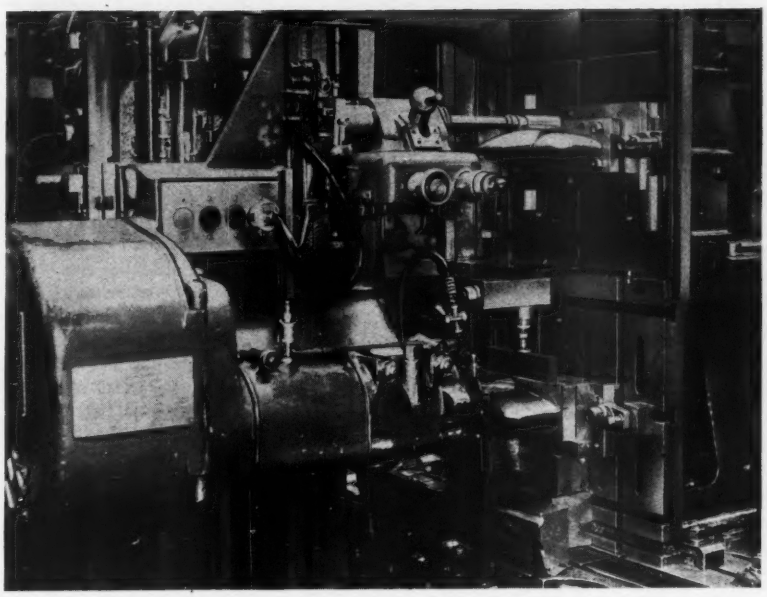


Fig. 1. Machining a press tool on a Keller die-sinker.

(1) Laboratory, serving all sections. This division has very much wider terms of reference than is usual for for a works laboratory. It is responsible for process checking as well as product testing and research.
(2) Tool room, serving all sections.
(3) Machine shop, serving all sections.
(4) Heat treatment department, serving all sections other than the bumper division.
(5) Bumper division.
(6) General press shop for lock and window regulator components.
(7) Automatic and manual plating plants dealing with components for locks, handles, window regulators, etc.
(8) Assembly sections for locks, handles and window regulators.
(9) Plastic and steering wheel division which is housed in a separate factory.

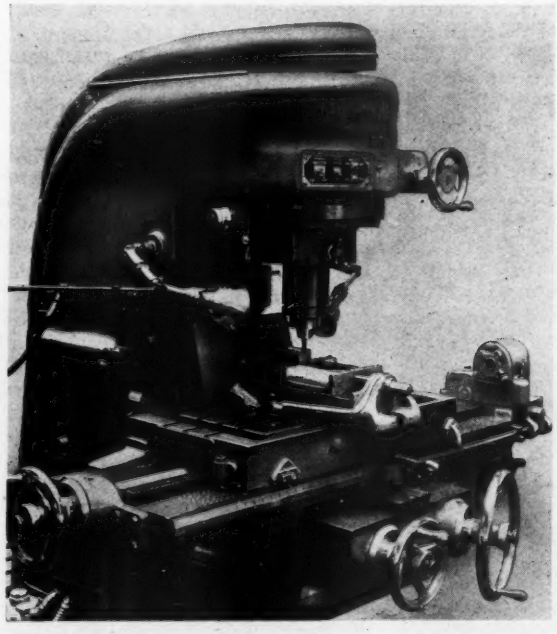


Fig. 2. Machining a mould for plastics on a Kearney and Trecker rotary head die milling machine.

There is also a large process planning and tool design section that plays a very important part in the organization of the flow of work through the factory. This section works in close liaison with both the design department and the various production departments. Because of the diversity and multiplicity of products, such planning is a key function. The planning department scrutinizes every new design to see that the maximum possible degree of standardization is obtained on components for such items as locks and window regulators. Such standardization is of importance for the economic production of work on presses and automatic machines. The planning department is also responsible for obtaining the best possible machine loading, not an easy matter when many varieties of parts are needed in widely differing quantities.

The tool room

The chief functions of the tool room are the production of press tools for the bumper division and the general press shop, tools, jigs and fixtures for the machine shop and the production of moulds for the plastics division. In the main these tools are produced by conventional methods, but the

Fig. 3. A muffle furnace for heating rolled bumper bars preparatory to forming the end.

machining is carried out in such a manner and with such precision that only a minimum of hand finishing is required. This is an important factor in the maintenance of delivery dates for tools and, of course, in keeping tool costs to a minimum consistent with high accuracy.

Dies are produced on Keller and Pratt and Whitney die-sinking machines, as are the larger and more elaborate punches. A typical set-up on a Keller machine is shown in Fig. 1. Small punches of relatively simple form are produced on punch shapers. Many of the moulds for the plastics divisions are produced on a Kearney and Trecker rotary head die milling machine. A mould for the production of small components may have as many as 48 impressions, and on the Kearney and Trecker machine, shown in Fig. 2, they can all be produced without any marking-out.

The tool room equipment includes a number of try-out presses on which all dies, except the largest used in bumper bar manufacture, can be "bedded-in" and thoroughly tested before issue to the production department. Large dies must be tried out in the production presses.

Bumper manufacture

Bumper bars are produced by three methods, rolling, pressing and hydraulic forming. Rolling and pressing are methods adopted by all bumper manufacturers, but hydraulic forming is a process that was originated by Wilmot Breeden Ltd. and to the best of our knowledge it is not employed by any other organization. Certainly, the actual hydraulic forming machines, developed in conjunction with Redman Tool Products Ltd., are unique.

In general, rolled bumper bars are made from hot-rolled 0·36–0·41 carbon steel. They are usually of fairly simple cross-section. The material is received as coiled strip and is first de-coiled and flattened and then cropped approximately to length. After being cropped the strips are transferred to a Rhodes press in which they are cut to correct length and end profile.

Rolling to form is carried out in one of a battery of machines with six pairs of forming tools to produce the desired cross-section. The rolls are progressive and must be carefully designed to ensure that the material is properly worked during its progress through them. These six pairs of forming rolls merely produce the desired cross-section. Therefore before the work leaves the rolling machine it passes through a five-roller bending unit to be given the correct sweep. Some types of bumper bars have a tendency to twist as they pass clear of the rolls, and to counteract this tendency the rolling machine incorporates an anti-twisting roller at the exit end.

From the rolling machine the bar is transferred to a press in which all the necessary holes are pierced. At this stage the procedure may vary as between one design of rolled bumper bar and another. All bumper bars have an end radius that is much less than the radius of the sweep, and the ends must be formed to radius in a press. On some designs cold pressing can be employed to form the ends, but on other designs the amount of work involved is too great to allow cold pressing. Such bumper bars are furnace heated and then hot-pressed to form.

The plant for heating and forming the ends comprises two muffle furnaces and two presses. A simple form of conveyor is used in conjunction with each furnace, see Fig. 3. In the first furnace one end of the bar is heated and then raised in the adjoining press. The bar is then loaded on to the conveyor for the second furnace in which the other end is heated. Incidentally, the conveyor for each furnace is loaded manually, but at the end of the travel through the furnace the bar is automatically removed from its carrier to fall down a chute to a position adjacent to the operating station for the press. After the end raising operations every bumper bar is checked against a form gauge and set, if necessary. It is then thoroughly de-greased and pickled and the attachment brackets are welded into position. The welds are then shot blast and the

Fig. 4. Tooling for drawing Vanguard bumper corner bars on a Wilkins and Mitchell 700 tons press.

bar is ready for the polishing and plating sequence.

Pressed bumper bars are of more complex form than rolled bars and have a more complicated operation sequence. They are made from fully-finished, deep drawing quality medium carbon steel. The production methods employed may be illustrated by a consideration of those used in the manufacture of corner bars for Standard Vanguard bumpers.

The material for pressed bumper bars is received in sheet form and is cut to blank length in a guillotine. At the first press operation the guillotined blank is drawn to form in a Wilkins and Mitchell 700 tons press fitted with an air cushion. To ensure a balanced proportion of right and left hand front and rear corner bars two sets of tools are mounted in the press together, see Fig. 4. One set produces a pair of front corner bars as a single pressing while the other set produces a pair of rear corner bars, also as a single pressing. Incidentally the simultaneous production of front and rear bars on one press is applied at all press operations except two. Apart from these exceptions the presses are arranged in line and the tooling is so arranged that front bars are inserted and removed from one side of the machine and the rear bars from the other. In other words, all front bars progress down one side of the press line and rear bars down the other.

After the drawing operation the drawn shell is passed to the appropriate one of two rotary shears on which the excess metal is removed from the flange. For the next operation, four tools are mounted in a Wilkins and Mitchell 150-ton press. They comprise two clipping tools and two six-hole piercing tools. As before, the tools on one side of the press are for front and those on the other for rear corner bars. The operation cycle is, mount at the first station and clip, then transfer to the second tool for piercing. This second tool is arranged to pierce two vertical holes from the top tool and to pierce four horizontal holes from the side-operated tools.

At this stage the front and rear bars are dealt with on separate presses on which they are parted into single bars. The pairs of front and rear single bars are then brought together again to a press in which four tools are mounted. Two of these tools take right and left hand front bars and the other two take the right and left hand rear bars. The top flange is clipped and the side form joggled at the first station and the bottom flange is clipped at the second station. These operations are carried out on a Wilkins and Mitchell 700 tons press. The tooling is shown in Fig. 5.

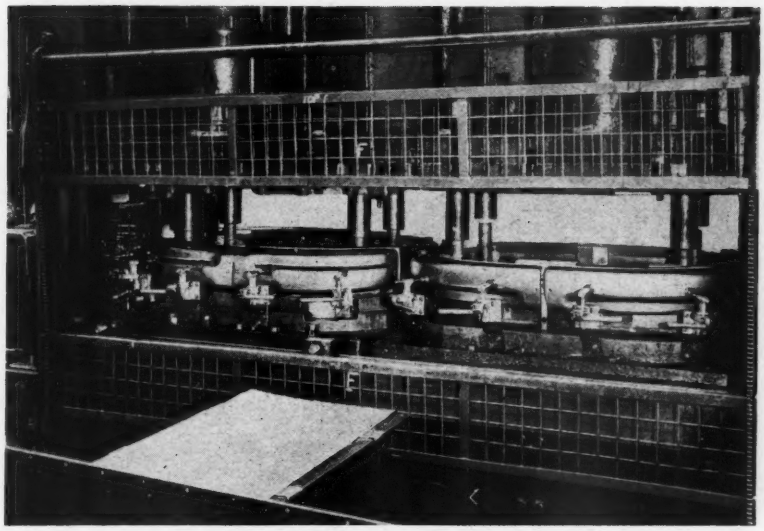


Fig. 5. Tooling for clipping the flanges of corner bars.

Once again the front and rear bars are transferred to separate presses tooled for end clipping, see Fig. 6. Two tools are mounted in each press, one for right hand and one for left hand bars. The tooling is so arranged that two operators work at each press, one from the front and one from the back. Side action tools are used. The front and rear bars then come together again in a 150-ton press for flanging. This completes the press operations on the front corner bars, but an additional operation is necessary on the rear corner bars. It is not feasible to produce the correct form at the ends of the rear bars during the first drawing operation, since the amount of cold working that would occur would have a tendency to cause fracture of the material. Therefore the correct end form is produced at a final press operation.

Centre bars for Vanguard bumpers are produced in three operations. The tools for these operations, six in all, are mounted together in one press. Three tools are for front and three for rear centre bars. The first tool clips the form for the top flange, the second performs a raising operation, and the third a joggling and flanging operation. This arrangement of tools allows a pair of centre bars to be produced at every stroke of the press.

As was stated earlier, Wilmot Breeden Ltd. have been pioneers in developing the hydraulic forming of bumper bars. This is a development of major importance. It does not supersede either the rolling or the press forming method, but it does make possible the production of shapes that could not be produced by the conventional methods. In other words, shapes that it would be difficult, if not impossible, to produce by rolling or press forming can be easily produced by hydraulic forming.

Fig. 6. End clipping Vanguard corner bars.

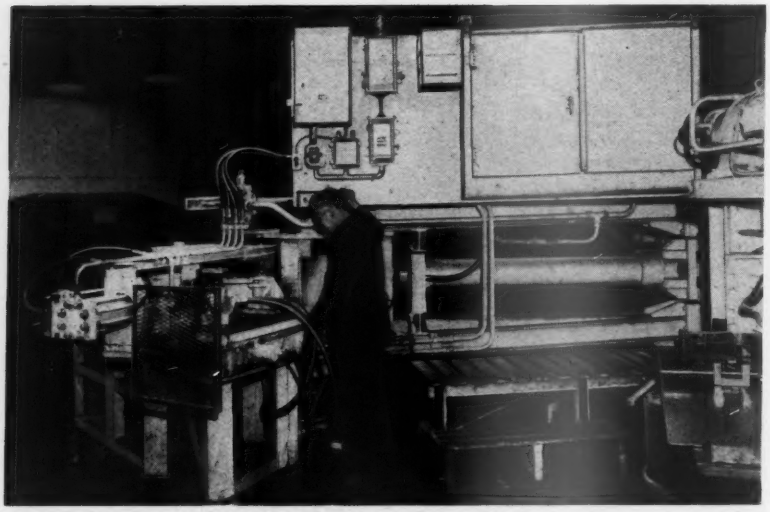


Fig. 7. Hydraulic forming machine for the production of bumper bars by a process originated by Wilmot Breeden Ltd.

Essentially, the hydraulic forming machine, which is illustrated in Fig. 7, comprises a pommel shaped to the internal form of the bumper bar and a slipper block with inserts to give the required external form. The slipper block is arranged to move in relation to the pommel in such a manner that at the end of its travel it has formed the metal to the correct shape for cross-section, sweep and end form. With this machine it is possible to produce the bumper bar direct from strip that has merely been blanked to the correct length and end form. In practice, however, it has been found advisable to pre-form the strip to a channel section before the final forming in the hydraulic machine. A pre-formed channel and a front and rear bumper bar produced on the machine are shown in Fig. 8.

The geometry of the slipper block movement and the mechanics of the hydraulic system are very complex. Here it must suffice to say that the work is positively located on the pommel at one end and is then clamped securely by hydraulic pressure. The slipper block is then advanced hydraulically and brought into contact with the work under a specific pressure. A large hydraulic cylinder then comes into action to cause the slipper block to pivot about a fixed point. A fourth hydraulic cylinder imparts a secondary pivoting motion to the slipper block to produce the end form and also ensures that the correct pressure between the slipper block and the work is maintained throughout the cycle. In addition, the hydraulic system is so controlled that the rate of movement of the slipper block is automatically varied to suit the great change in curvature that occurs towards the end of the bumper bar. This hydraulic forming machine has a completely automatic cycle.

Although it is not possible to discuss the design details of this unique machine, there is no doubt that it has remarkable potentialities in bumper bar manufacture. As already stated, it does make possible the production of forms that would not be practicable by any other method. It should however, be stressed that even for hydraulic forming, the simpler forms are preferable. This method can also lead to greatly reduced tool costs, since instead of several sets of expensive press tools, the only requirements are the relatively simple tools for producing the channel section and the formed pommel and the inserts for the slipper block. Finally, there is economy in material. The blank need be only the correct width, whereas in press forming the blank to be drawn must include an appreciable amount of surplus metal which is afterwards clipped off.

Polishing and plating bumper bars

The production of a durable and highly polished plated surface calls for the most exact control at every stage of the processing. To begin with the base metal surface must be given a very high degree of polish before any attempt is made at metal deposition. This is effected in two polishing operations, each carried out in two stages. In both the rough and fine polishing operations, the first stage consists in bobbing the sweep of the bar on an automatic polishing machine. At the second stage the ends are polished on hand machines. This polishing is extremely important inasmuch as the better the finish on the base metal the greater the adhesion between the deposited nickel and the base metal surface. Furthermore, the better the polished surface on the base metal, the easier it is to produce the specified final brilliant plated surface.

At present all bumper bars are plated in a manual plant, by Wilmot Breeden Ltd. are shortly expecting delivery of an automatic plant for both nickel and chrome plating. When the bumper bars are received from the polishing department they are mounted on jigs and then pass through the following processing sequence :

(1) Electrolytic cleaning in a hot alkaline solution.
(2) Rinse in cold running water.
(3) Electrolytic etching in a solution of sulphuric acid.
(4) Rinse in cold running water.
(5) Scour to ensure an absolutely clean condition.
(6) Rinse in cold running water.
(7) Heavy nickel plating to obtain a minimum thickness of 0·00125 in.
(8) Rinse in cold running water.
(9) Rinse in hot water.

To obtain the specified minimum thickness of nickel at the point of least deposition, it is generally necessary to have an average deposit of 0·002 in. or more. The area of the anodes and the positions of the anodes in the vat are determined to suit the area and form of the bars to be treated, but even so the average deposit is always considerably more than the

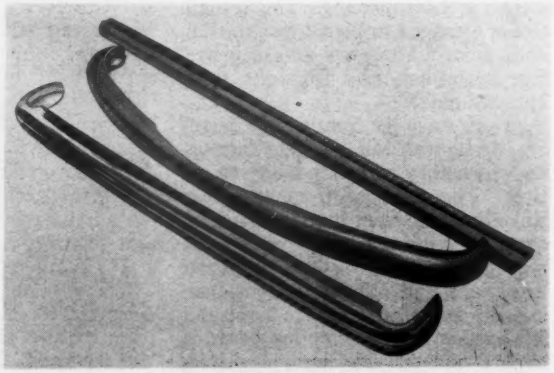


Fig. 8. Front and rear bumpers produced on the hydraulic forming machine. A pre-formed channel section is also shown.

specified minimum. Really good nickel plating is essential. Adequate thickness must be allied to very good adhesion if the final coating is to serve its protective purpose. Every possible care is therefore taken throughout the nickel plating sequence. The nickel solutions are agitated by warm air and continuously filtered by the most advanced technique. In addition, the solutions are frequently filtered into spare tanks to allow the plating vats to be thoroughly cleaned out. The tests carried out to ensure that the specified standards are maintained will be discussed later in connection with laboratory control.

From the nickel plating department the bumper bar is transferred to the polishing department for nickel finishing. This is important so far as the final appearance of the product is concerned. At present nickel finishing may be carried out in two ways, either by using twin-head automatic machines for the sweep followed by hand polishing for the ends, or the whole length of the bar may be polished in one pass through an eight-head straight-line polishing machine, see Fig. 9. On this machine each of the eight heads can be swivelled to any desired angle, and each head is counterbalanced in such a manner that as the mop follows the sweep of the bar a constant pressure is maintained. Provision is now being made for much greater use of multi-head straight-line automatic polishing machines.

When nickel finishing is completed the bars are transferred to the chrome plating section. At present this is also a manually-operated plant. The bars are first placed on jigs and then pass through the following processing sequence :

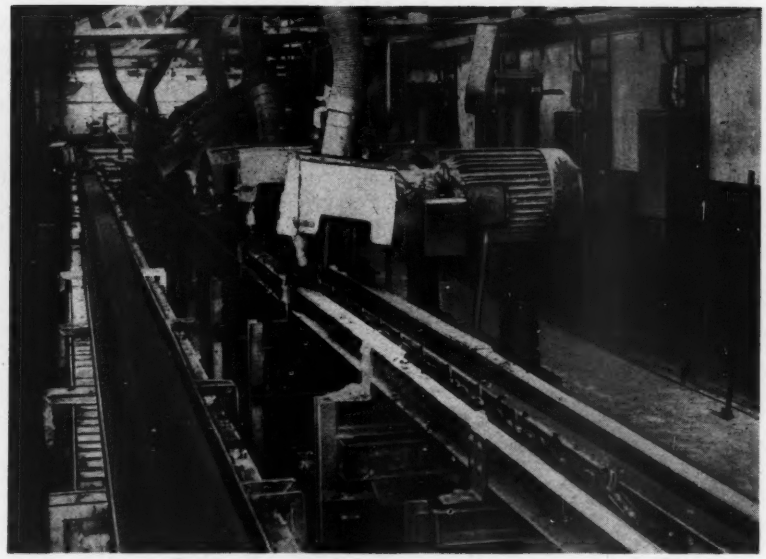


Fig. 9. Eight-head straight-line polishing machine for bumper bars.

(1) Chrome plate to a minimum thickness in the range 0·00001–0·00002 in.
(2) Rinse in water.
(3) Rinse in alkaline solution to neutralize any remaining chromic acid.
(4) Rinse in cold running water.
(5) Rinse in hot water.

At this stage the bumper bars are inspected and are then sprayed on the back surface with a special aluminium paint before they are passed to the assembly section. It should perhaps be stated that the procedure described is for passenger car bumper bars. For certain commercial vehicles the bumper bars are not plated but are given two coats of black enamel. Enamelling is carried out in a continuous, high-temperature conveyor plant which is air-conditioned to ensure a dust-free atmosphere.

Fig. 10. Discharge end of rotary hearth furnace for hardening bumper springs.

Bumper springs

Bumper springs are produced from 0·65–0·75 carbon steel. The operation sequence varies according to the design, but in general the desired form is produced in a series of hot and cold pressing operations, followed by hardening and tempering. For hardening, the springs are first raised to the required temperature in a large diameter rotary hearth furnace, see Fig. 10, that is loaded and unloaded manually. On unloading, the springs are placed on a chute down which they fall by gravity into an oil quench tank. The springs on leaving the chute, fall on to a conveyor that carries them through the tank and out to a position adjacent to the rotary hearth tempering furnace. After being tempered, every spring is barrelled to remove all furnace scale and is then set to a gauge. Before the springs are passed to the assembly section they are enamelled in the continuous plant used for commercial bumper bars.

Overiders

Overiders for bumpers are produced in the general press shop. They are made from deep drawing quality medium carbon steel. To begin the operations the overider is blanked from strip and is then drawn to form. Conventional tools are used, but it should be stressed that they have to be very carefully designed and made to produce a shell that is not only to the desired form but is also free from wrinkles or other surface defects,

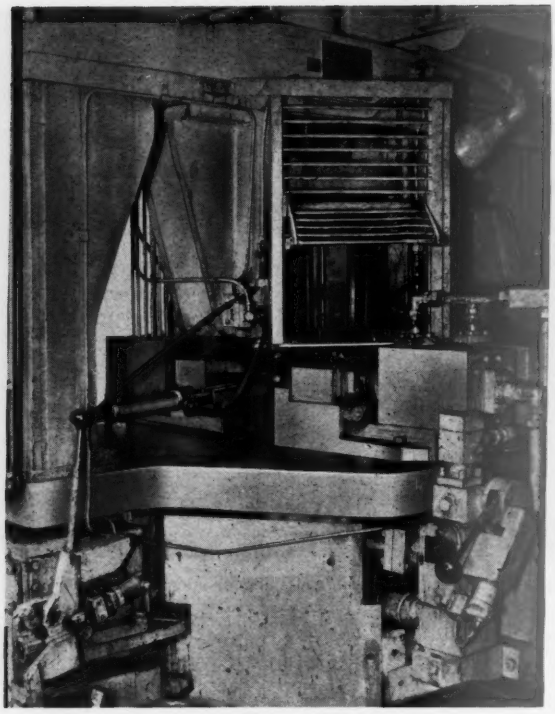


Fig. 11. One of the special multi-tool hydraulic presses for clipping bumper overiders.

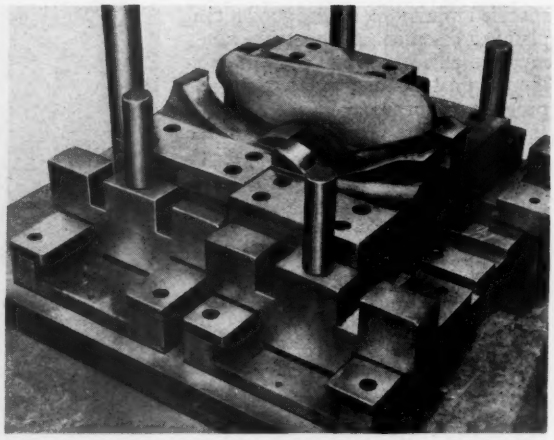


Fig. 12. The setting fixture for the pommel and tools used in the press shown in Fig. 11.

since quite minor defects will necessitate excessive polishing before the plating is carried out. With many of the overider forms it is not an easy matter to produce the necessary high quality of surface.

After the drawing operation the shells are transferred to special multi-tool hydraulic presses on which they are clipped to depth. One of these machines is shown in Fig. 11. The workpiece is mounted on a specially formed pommel and the clipping is effected by four slitting tools, two at the back and two at the front of machine. Fig. 12 shows the setting the fixture for the pommel and tools. Tool movements are operated by an hydraulic system which is so arranged that diagonally-opposed tools move in simultaneously. An interesting addition has been made to this press for removing the metal clipped off the shell. At each side of the press there is an air cylinder with a tong shaped extension attached to the piston rod. Each extension reaches to the inside of the drawn shell on the pommel. After the four clipping tools have advanced and retracted, the air cylinders come into action automatically to withdraw the extension pieces. As each one is withdrawn it brings with it a half clipping.

Considerable cold working occurs during the deep drawing operation. Any further press work on the material in the work-hardened condition would probably lead to splitting, so at this stage the drawn and clipped shell is bright annealed in a continuous Birlec controlled-atmosphere furnace of the type shown in Fig. 13. The final press operation is to planish and form the flange on a coining press. This completes the major operations on the overider except for the polishing and plating. For some overiders the polishing and plating sequence is the same as that described for bumper bars, but for others bright nickel plating is used. Where bright nickel plating is used, the nickel finishing operations are eliminated, but an even higher degree of finish must be given to the base material before the nickel deposition sequence.

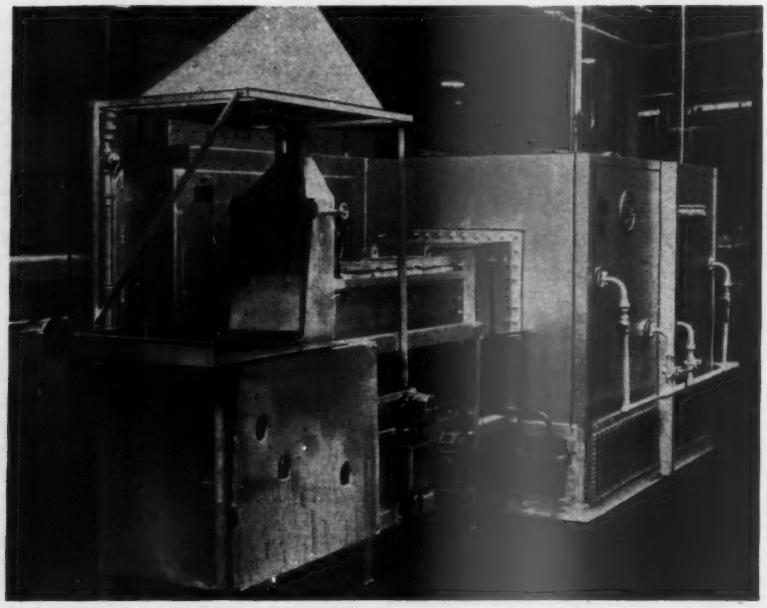


Fig. 13. One of the battery of Birlec controlled atmosphere furnaces.

Automatic Plating

It is not intended to discuss in detail the manufacture of handles, but merely to describe some of the more important and interesting operations. Handles and their escutcheons, as well as certain locks, entail the use of zinc-base die castings which must be plated to give a protective and decorative finish. Naturally, the die castings are first thoroughly polished to a high surface finish. Where the

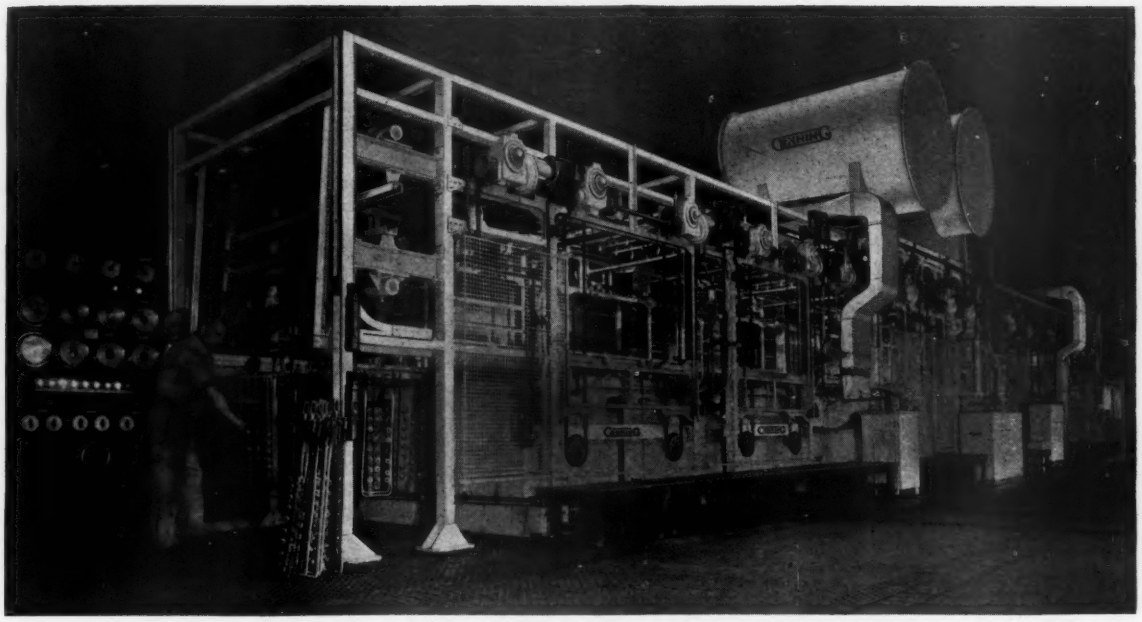


Fig. 14. One of the automatic copper-bright nickel-chrome plating plants.

quantities of any one type are sufficiently large, automatic polishing machines are used. After the polishing is completed the die castings are transferred to the plating section, which comprises two fully-automatic copper-nickel-chrome machines, each 95 ft. long. There the work passes through the following processing sequence:

(1) The work is loaded into baskets and put through an automatic trichlorethylene de-greasing plant.
(2) De-greased components are placed in jigs or wired on frames according to their shape and size, and put on a conveyor for transfer to the loading end of the automatic plating plant.
(3) The jigs or frames are transferred to the automatic plating plant conveyor to be carried through the following :
(4) Alkaline cleaning.
(5) Rinse in cold water.
(6) Dip in dilute hydrofluoric acid.
(7) Rinse in cold running water.
(8) Dip in dilute cyanide.
(9) Bright copper plating.
(10) Rinse in cold running water.
(11) Dip in dilute sulphuric acid.
(12) Bright nickel plating.
(13) Rinse in cold running water.
(14) Inspection bay.
(15) Chrome plating.

Successful operation of an automatic plating plant is possible only if the strictest control is exercised over a number of factors. For example, the current density is calculated to suit the area to be plated and must be maintained within very close limits. Even closer control must be maintained on the contents of the copper and nickel vats. To maintain the very high degree of purity that is necessary, high speed, continuous filtration is employed in each vat. In addition, both the copper and bright nickel solutions are continuously air-agitated. Twice a week the contents of both vats are filtered into spare tanks so that the plating vats can be thoroughly cleaned. Another important factor in maintaining the purity of the bright nickel solution is that the solution is in continuous circulation from the plating vat to a separate tank and back again. The separate tank is equipped to give continuous electrolytic low-current density purification of the solution.

The specified minimum thickness for copper and bright nickel is dependent upon whether the components are for external use with complete exposure to the weather, or for

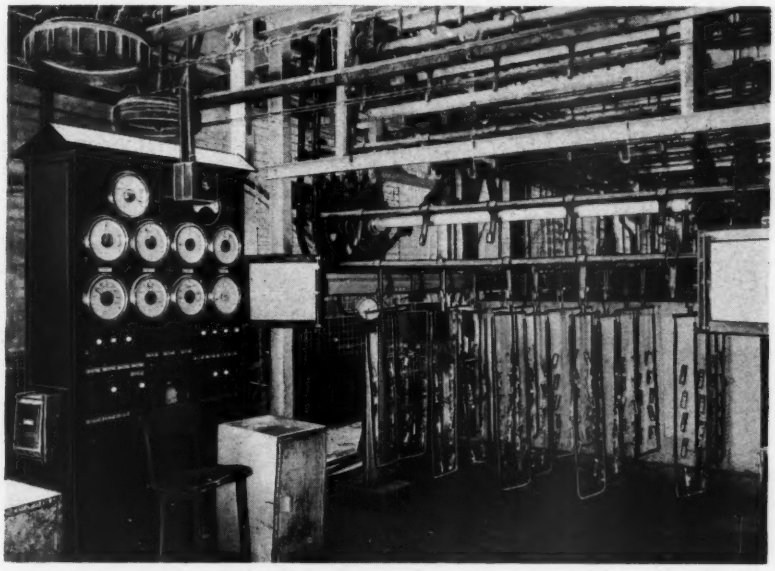


Fig. 15. The control panel and the loading station of an automatic plating plant.

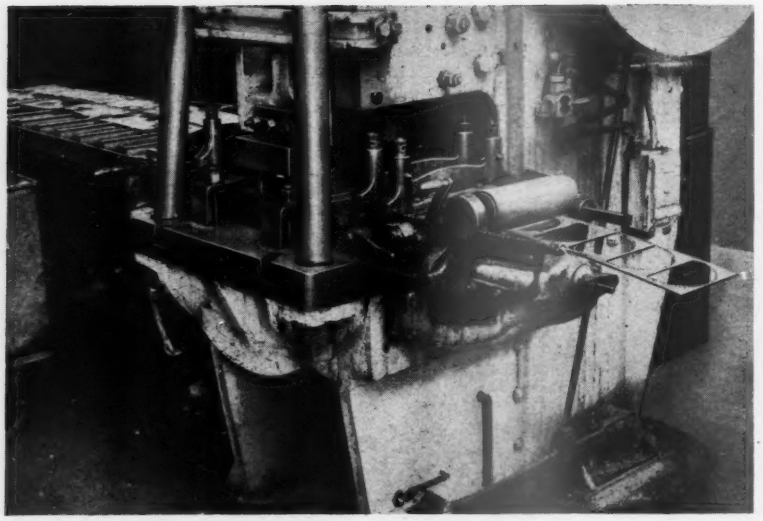


Fig. 16. Press with automatic feed for blanking window regulator segments.

interior use with some measure of protection. For the bright copper plating the specified minimum is 0·0003–0·0004 in., while the combined copper and nickel must not be less than 0·0006 in. for interior work and 0·0008 in. for external use. The minimum for chromium plating is 0·00001 in. To maintain these minima, a much heavier average thickness is required. One of the automatic plating plants is illustrated in Figs. 14 and 15.

The work carrying fixtures used in the automatic plating plant are returned to the loading station by a conveyor that carries them through a series of tanks with contents that ensure that the jigs or frames are completely free from any carry-over from the chromium vat. This is essential since the bright copper plating solution is highly susceptible to contamination by chromium.

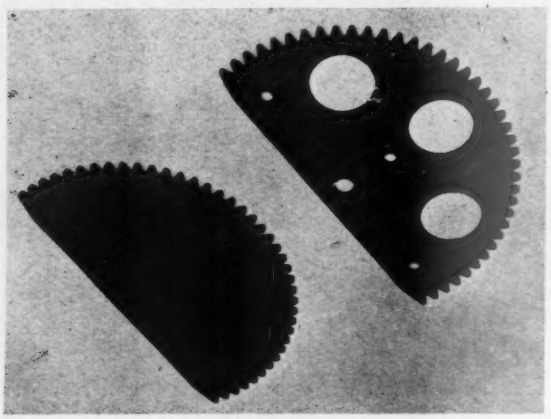


Fig. 17. A blank as produced on the press shown in Fig. 16 and a finished segment.

Locks and window regulators

As in the case of handles, it is not intended to follow through the manufacture of all the components for locks and window regulators, as brief descriptions of selected operations will suffice to illustrate the methods employed. In general, all other than bought-out parts for these assemblies are produced either in the machine shop or in the general press shop. Most of the work in both these departments is conventional in character and therefore call for little comment. It may however be pointed out that in the machine shop remarkably high output rates are obtained from single spindle and multi-spindle automatics on a very wide range of components.

One of the more interesting operation sequences in the general press shop is that used in the production of toothed segments for window regulators. At the first operation, strip is fed automatically into the press shown in Fig. 16 and the form, including the teeth is blanked so accurately that a shaving operation on another press is sufficient to produce teeth of the requisite accuracy. A blank and a finished segment are shown in Fig. 17. Reference may also be made to the production of small seven tooth pinions for window regulators. These are produced complete in two operations. At the first, which is carried out in the machine shop, the blanks, turned, bored and spigoted, are produced on a bar automatic. The teeth are produced in a single stroke on a press which forces the work through four progressive broaching dies to produce teeth of quite accurate form.

Reference may also be made to the production of cams for push button locks. To give satisfactory performance these cams must have accurate registering surfaces. They are produced by surface broaching on a Weatherley Oilgear single slide machine illustrated in Figs. 18 and 19. One push button lock incorporates three cams, one of which must be machined on two surfaces. The Weatherley machine is shown tooled for these cams. An indexing type work-holding fixture is mounted on the shuttle table and is arranged to take twenty cams in four stacks of five, that is, there are ten of one form of cam and five of each of the other forms. This arrangement allows fifteen cams to be finished in a single pass of the broach, that is, every stroke of the machine produces cams for five locks. Similar surface broaching methods are used for some window regulator segments.

It is not necessary to discuss in detail the methods employed for the production of lock and window regulator sub-assemblies, but it may be stressed that there is complete equipment for making every desired type of fastening. For example, considerable use is made of copper brazing to reinforce interference fits. In every suitable application this type of brazing is carried in a Birlec continuous, controlled-atmosphere furnace. Provision is also made for high-frequency brazing, and for spot, projection, flash-butt and arc welding, so that the most appropriate method can be used for every application.

A complete reorganization of assembly methods for locks has recently been carried out, particularly for the types required in large quantities. These new assembly lines have been planned to serve three purposes :

(*a*) to give increased output with greater accuracy from the same labour force.

(*b*) to make the best possible use of the available floor area ;

(*c*) by careful design and a measure of standardization to reduce the stock of component parts hitherto considered necessary.

The new assembly lay-out will allow greatly increased output to be obtained without imposing additional strain on the operators. In fact much less physical effort will be required. This is chiefly due to the fact that Broom

and Wade and other types of pneumatic squeeze riveters have been substituted for the fly presses previously used. These riveters are arranged for pedal operation so that the operator always has both hands free for manipulation of the work. The components needed at any station are stocked in specially designed flow containers fixed to the bench in positions convenient for the operator. One of these lines is shown in Figs. 20 and 21, and a handle assembly line in Fig. 22.

Plastics division

So far as the automobile industry is concerned, steering wheels are the main products of the plastics division In addition, there is a substantial production of knobs for handles and gear levers and of push buttons for the various dashboard controls. There is also production of three-dimensional plastic badges. These are not in any way functional, but are purely decorative.

All the operations for steering wheel production are carried out in the plastics division. To start the manufacture of a steering wheel rim, wire of the appropriate diameter is cut to length and then rolled to form a ring of a diameter some $\frac{5}{16}$ in. less than that finally required. The ends of the ring are then welded together in a semi-automatic butt welding machine. After the welding operation, the rim is mounted on what is in effect an expanding mandrel in a circular hydraulic stretching press to be expanded to the correct diameter. This operation not only produces a truly circular rim of the correct diameter but also acts as a check on the efficiency of the weld, since the force required to increase the diameter by $\frac{5}{16}$ in. is large enough to impose considerable stress at the weld.

Meanwhile work is also proceeding on the hub. Hub castings of high accuracy are supplied by an associated company but any necessary machining of the splines or taper bores is carried out in the department. Some steering wheels have hubs with cast-in spokes, but most Wilmot Breeden steering wheels have inserted spokes that are an interference fit in the hub. A special three-head hydraulic machine has been developed for inserting all the spokes simultaneously to a controlled length. It is illustrated in Fig. 23. Incidentally, experience has shown that interference fit spokes inserted on this machine are more secure against loosening than spokes cast into the hub.

At this stage the spoke and hub assembly is welded to the rim, and after being welded the complete assembly is checked for flatness and concentricity, and if necessary, corrected. The spokes are then masked, and the hub, after being treated in a phosphating solution to improve adhesion, is sprayed with the appropriate paint. The wheel is then ready to have the plastic rim moulded on to it. Both injection and compression moulding are used in the production of Wilmot Breeden steering wheels. In general, injection moulding is used for wheels required in large quantities and compression moulding for wheels required in only relatively small batches.

Various makes of injection moulding machines are installed, but all steering wheel production by injection moulding is carried out on Lester machines of the type shown in Fig. 24. On this machine an accurately metered amount of powder is fed to the heating system and is then injected under great pressure into the dies, which have a closing pressure of 750 tons. The machine cycle is fully automatic and is completed in less than one minute. There is scarcely any flash at the joint and final dressing is much easier than with a compression moulded wheel. In sum, it may be said that for large quantities injection moulding is much quicker and much more economical in material than compression moulding.

A recent plastic development that is, we believe, carried out only by Wilmot Breeden Ltd. and one other organization in this country, is the process known as three-dimensional plastic moulding. It is used mainly for the production of badges and escutcheons. By this technique an almost unlimited combination of metallic and colour finishes is possible. The final product is more colourful,

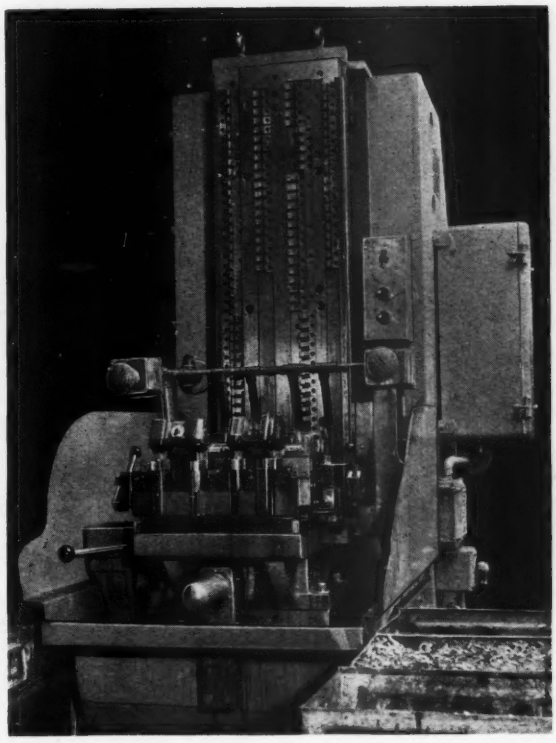


Fig. 18. Weatherley Oilgear surface broaching machine for broaching cams for locks.

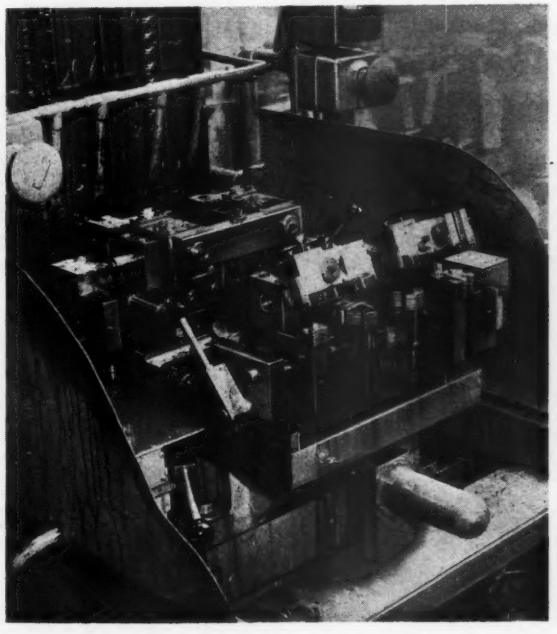


Fig. 19. The work-holding fixture on the machine shown in Fig. 18.

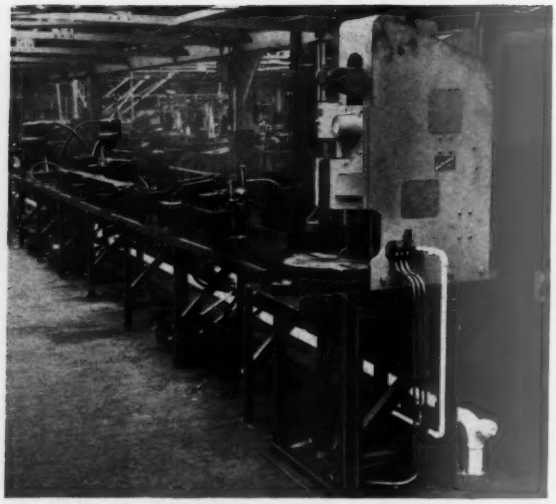


Fig. 20. An assembly line for push button locks.

attractive and permanent than similar kinds of products made by other methods.

Normal injection moulding practice is employed, but the conditions are more critical, since to obtain the desired effect there must be complete freedom from flow lines and sinking, and there must be absolute clarity. Paints are applied by standard spraying techniques, but there are several methods of metal deposition. Wilmot Breeden Ltd. employ the vacuum deposition method in which the part to be treated and the metal for deposition are mounted in a vacuum chamber. Provision is made for heating the metal until it finally disintegrates into molecules that fly in all directions. The work is masked except for the surfaces to be treated, and the metal molecules bombard, and adhere to, the exposed plastic surfaces.

The laboratory

Wilmot Breeden Ltd. have a much larger laboratory organization than is common for engineering firms of comparable size. In addition, the terms of reference for laboratory activities are much wider than in most engineering laboratories. Naturally, there is day-to-day control of routine activities, but in addition research and development projects on many different subjects are also undertaken. Because of the wide range of materials and products involved, it has been found advisable to divide the laboratory into several sections under the general direction of the laboratory manager. Each section is under a specialist in the particular subject with which the section is concerned.

A brief resumé of the work of the various sections will give some conception of the laboratory activities. There are three chemistry sections. One deals with routine chemical analysis for testing and controlling plating solutions and associated processes. The second is concerned with organic chemistry, connected particularly with the examination of paints, lacquers, oils, thinners and greases, while the third is a research unit particularly concerned with plating problems. Metallurgical matters are

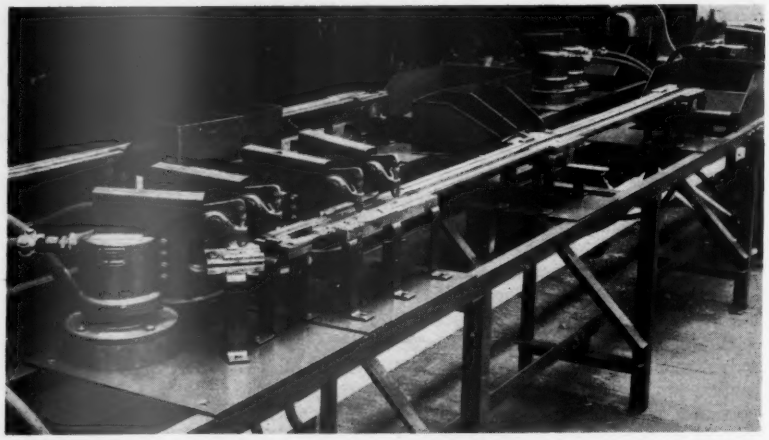


Fig. 21. Part of the line shown in Fig. 20.

Fig. 22. Part of the handle assembly section.

dealt with by two sections, one dealing with the analysis of steel and non-ferrous metal supplies, while the other is concerned with day-to-day problems and with special investigations into fabricating techniques in with the metallurgical factor may be important. Prototype components and revised designs of components are thoroughly tested in the engineering section of the laboratory before production is sanctioned. A recently constituted section is exploring the possibilities of applying electronic techniques in some of the activities of the Company. There is also a library and a photographic section. There is a well-stocked technical library and in addition the librarian controls an abstracting service to provide the laboratory staff and the factory staff with information about new developments. A technical abstract bulletin is issued monthly.

Because of the important part the

laboratory organization plays in the maintenance of product quality and in the development of improved techniques, it is worth describing some of the many activities in fuller detail. For example, the routine chemical analysis section maintains a daily control on all plating and other chemical and metal finishing processes. On the nickel plating vats for both bumper bars and die-cast parts, a measurement of the pH value is made daily. This involves tests on 25 vats for bumper bars, two automatic plants and 31 manual vats for die-cast parts.

Twice weekly determinations are made of the free cyanide, copper content and sodium hydroxide content of the copper vats in the automatic plants and the manual plants for plating zinc-base die castings. Once a fortnight the nickel vats for bumper bars are analysed for nickel sulphate, nickel chloride, nickel formate and boric acid. Similar analyses are made for bright nickel solutions with additional analyses for sodium formate, cobalt sulphate and formaldehyde.

Factors other than the actual plating solutions can and do affect the quality of plating, and a further section of the laboratory carries out regular checks on the quality of the work produced in order to ensure that any of these other factors are acting to the detriment of quality. This involves the measurement of plating thickness with reference to standard specifications and the examination of certain plated components under the salt spray porosity test.

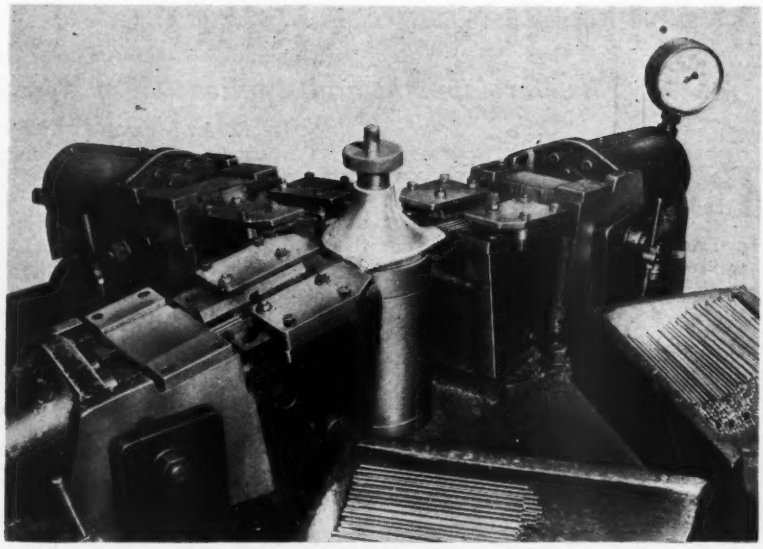


Fig. 23. Special three-head hydraulic machine for inserting steering wheel spokes into the hub casting.

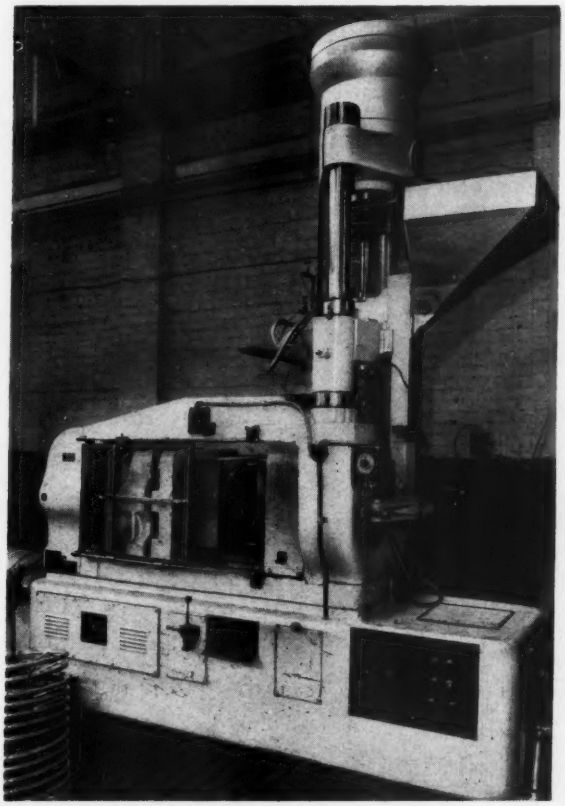


Fig. 24. A Lester injection moulding machine for steering wheels.

Jet spray thickness tests are carried out on samples of every different type of zinc-base die casting plated during the day and the night to determine copper, nickel and chromium thicknesses. A more searching test but one that for physical reasons must be carried out on a smaller number of components is the salt spray test. For this test, the samples are left in the salt spray room for one week. A 20 per cent salt solution is sprayed in the form of a very fine mist over the samples for eight hours in every twenty-four. The samples are viewed every day and examined for deterioration, as shown by the number of spots of corrosion that have appeared. It has been found possible to adopt a standard that sets a maximum for the number of spots of corrosion per unit surface area for acceptable work.

The engineering and mechanical testing section of the laboratory is responsible for a wide range of activities that have considerable influence on product quality. In addition to tests on prototype mechanisms, work is also carried out of a more fundamental nature, including such items as the determination of frictional characteristics of certain lubricants and greases, the study of car internal lighting and the measurement of noise associated with car auxiliary components. There is also periodical testing on production locks and window regulators under conditions that simulate those that will be encountered in service. This section has not only to carry out the actual tests but has also in most cases to design and construct the testing equipment.

It is not possible to describe the work of the research section. However, it may be pointed out that while the other sections are chiefly concerned with the maintenance of quality on established products, the research section is engaged on more fundamental work of importance for the future.

Although reference has not been made to the works inspection system, it should be stressed that the laboratory work for control of quality is reinforced by the work of an organized works inspection staff. In fact, one of the outstanding points of the whole Wilmot Breeden organization is the manner in which every possible effort is made to ensure that the specified high standard of product is maintained.

MODERN CARBON COMPOSITES

The Development and Application of New Types of Material

By S. A. Murray*

DURING the past twenty years fabricated carbon and metal-carbon composites have become key materials in a wide variety of engineering applications. For certain specialized purposes demanding materials possessing unusual properties there is no equivalent alternative. This, of course, is quite apart from the electrical field which provided the initial major use of carbon, namely for current collection in electrical machines. Both in the form of amorphous carbon or in the form of graphite, carbon embodies a unique combination of physical and mechanical, chemical and electrical properties which have resulted in a wide diversity of applications. The grade of carbon used, and to some extent its precise manufacturing process, is largely determined by the purpose for which it is required, often necessitating some variation of the above properties. These can, however, be accurately controlled over a wide range.

The major properties of carbon from a mechanical viewpoint are :—

(1) Self-lubricating by virtue of the included graphite with its well known multi-layer structure.
(2) High resistance to temperature and thermal shock.
(3) Availability in impervious form.
(4) Machinability to a high degree of accuracy.
(5) It is not wetted by molten metals.
(6) High resistance to wear.

The element carbon remains solid in inert atmosphere up to very high temperatures, passing from the solid to a vapour at 3500/4000 deg. C. Having therefore no liquid phase it cannot be cast like metals. Thus carbon articles are usually fabricated from powder, agglomerated with a suitable binder. Selected forms of carbon are blended together and pitch, tar and other hydro-carbon binders are added. This mixture is compressed under very high pressures in dies or moulds to desired shapes. For some repetitive processes the mixture is made suitable for extruding into lengths of simple profile. Finally the compact is subjected to a lengthy heat treatment in a protective atmosphere at very high temperatures.

Regarding the composite metal-carbons, there are three standard Morganite metal-carbons, the first incorporating copper, the second white metal, and the third both copper and lead. Plain carbon is suitable for working in oxidizing atmospheres up to 450 deg C. Both the copper-carbon and the copper lead-carbon should be limited to 350/400 deg C., and the white metal-carbon to 150 deg. C., under similar conditions. In neutral or inert atmospheres the temperatures for carbon can be very substantially increased for a comparable wear rate. Plain carbons are inert to practically all chemicals other than concentrated oxidizing acids, but when for strength reasons a metal carbon component is necessary, the effect of a chemical on the metal content is a further factor for consideration.

The substantial mechanical benefits obtained by combining the self-lubricating properties of carbon with the added strength of metals is shown in the attached table. It is however possible to manufacture non-metallic carbons of considerably greater strength than that given, approaching in fact the metal carbon composites. All available field experience nevertheless substantiates the conclusions based on extensive laboratory investigation, that the metal-carbons exhibit a greater measure of consistency and are less sensitive to changes in operating conditions, *e.g.*, the variation in hardness of the mating member.

Regarding specific applications, primarily of the metal-carbons, the following are of interest to engineers in the automobile and allied industries.

Clutch Release Bearings

For many years the well known Borg & Beck clutches have employed on the majority of car applications a carbon thrust release bearing mounted in a C.I. cup. In recent years two major factors have contributed to revealing the need for a more robust and lower wear rate material ; (a) the use of private car engines in small commercial vehicles, and (b) the export market. In both cases the duty of the clutch has become more onerous than normally experienced in the pre-war market, for a variety of reasons. Among these is the tendency, following developments elsewhere in the unit, to use smaller clutches for the same torque capacity. This occasions increased unit loading on the release bearing, although this has been offset

*The Morgan Crucible Company Limited.

Morganite carbon gland rings for turbines and compressors.

Morganite metal-carbon and carbon bearings.

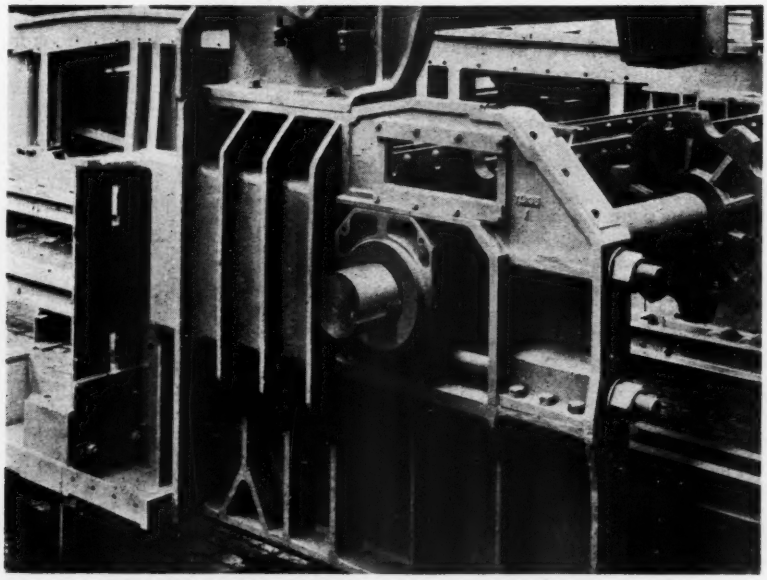
Table I

Grade	Sclero-scope hardness	Relative density	Approximate ultimate strength p.s.i.				Modulus of elasticity E (p.s.i.)	Coefficient of thermal expansion per °C.	Remarks
			Transverse	Compressive	Shear	Tensile			
CY1	55	1·65	3500	9000	2500	1500	$1{\cdot}1\times10^{6}$	$3{\cdot}4\times10^{-6}$	Plain graphite carbon
MY3D	70	3	13000	44000	8000	9000	$2{\cdot}9\times10^{6}$	$5{\cdot}2\times10^{-6}$	Metal carbon

in some instances by using a larger bearing. The transfer from right to left hand drive in the case of some of our smaller cars for export has further increased the burden of the release bearing by encouraging pedal riding from physical limitations of space.

Under pedal riding conditions spot temperatures of well over 1000 deg C. may occur at the mating surfaces of the bearing and the release lever plate. If sustained, this will initiate a partial breakdown of the bearing surface due to oxidation, resulting in a high wear rate. Metal carbon has a thermal conductivity approximately ten times that of plain carbon, thus effectively limiting these spot temperature conditions. In addition metal carbon exhibits a much greater insensitivity to the other effects of these pedal riding temperatures, *i.e.*, the appreciable softening of the release lever plate. Actual measurement has shown this "let down" to be as much as 50 per cent on an initial plate hardness of approximately 500 V.P.N. Since the wear rate of carbons is allied to the hardness of the mating member, there is no doubt that this has been a contributory factor to the failure of plain carbon under onerous conditions. Extensive bench and field experience over the past four years has proved the marked superiority of the metal-carbon thrust bearing over the plain carbon original, and it has now been adopted by some of the leading manufacturers in this country.

Initial field proving was undertaken on known arduous applications, *i.e.*, motoring schools, police cars and light commercial vehicles. Typical of these arduous conditions was an 18 h.p. Wolseley police car, using a 9 in. Borg & Beck clutch giving a unit loading of 120 lb. per sq. in. on the metal-carbon release bearing. Over a total mileage run of 68,000 the wear rate of the bearing was only 0·0011 in./1000 miles, *i.e.*, a total life in this particular size of bearing, having a maximum wear depth of 0·2 in., of approximately 190,000 miles. Such a low wear rate substantially reduces the need for clearance adjustment, the facing wear being compensatory. Further direct comparisons between metal-carbon and ball thrust races were made and are in progress, with every indication that the more costly alternative is unnecessary. All available data substantiate the view that the modern trend of lightly hydraulically loading the release bearing to obviate adjustment will not impair the performance of the new material.



Babcock & Wilcox chain grate stoker showing metal-carbon bearing.

Plain Bearings

Metal-carbon bearings of the journal type have been used successfully in a variety of applications. The dry friction of these bearings in conjunction with steel journals is of the order of 0·12, a high figure when compared with oil lubricated types. Under these dry conditions the load and speed limitations are necessarily low owing to the relatively high coefficient of friction and lack of coolant. A dry-running arduous condition satisfactorily met is that in the Babcock & Wilcox travelling grate stoker. These large diameter bearings up to 8 in. bore operate under loads of up to 350 lb. per sq. in. in high temperatures, which precluded the use of normally lubricated bearings. Cast iron bearings previously used could not be adequately lubricated, so that the rate of wear was high and the maintenance costly.

Although the performance of these large metal carbon bearings is of no direct interest to automotive engineers, the very low wear rate under these arduous, heavily loaded conditions *i.e.*, 0·002 in. per annum over seven years, is indicative of the material's capacity. By contrast small, lightly loaded carbon bearings are a standardized feature on the butterfly valve spindles of Zenith carburetters. Under wet operating conditions, as in fuel pumps for aircraft, when adequate coolant of negligible lubricating properties is available, the friction is reduced to the order of 0·001. Such low friction values obviously mean fluid film conditions, the self-lubricating properties of the metal carbon bearings affording safeguard against serious damage should for any reason the film condition break down. A particular fuel pump handling paraffin in which the metal carbon bearings are immersed, imposes journal loads of some 1,250 lb. per sq. in. at a peripheral speed of 850 f.p.m.

The well-known Lucas fuel pump fitted to the majority of British turbine engines has upper and lower bearings for the main rotor in Morganite metal-carbon material, the metals in this case being copper and lead. The unique chemical and self-lubricating properties have also enabled these bearings to meet the demands of "Bonderizing" plants. Here the bearings pass from one severe condition to another, as in the degreasing, stoving, cooling, hot water spray, chromatic rinsing and final drying at 450 deg F.

Pump and Shaft Fluid Seals

The self-lubricating property of carbons and metal-carbons is also important in their use for both face type pump seals and rotary shaft seals, also styled mechanical seals. Here, by virtue of their greater reliability they have replaced the older type of soft packing glands. Much ingenuity has been applied to developing various designs incorporating carbon materials as the sealing member. In some designs a rubber member is used for preventing axial leakage, while the carbon component seals radially, using the pump body or impeller as its counterface, or in some cases a collar on the shaft. Pressures of several hundred lb. per sq. in. can be handled, depending on the method of application.

A simple design developed for circulating water pumps for cars employs a specially shaped rubber moulding which is bonded to the carbon ring. This type of seal has also been used in aircraft control gear where the fluid is a thin oil, using a special rubber body in conjunction with copper/lead-carbon ring, limiting pressures in this case being 10 lb. per sq. in.

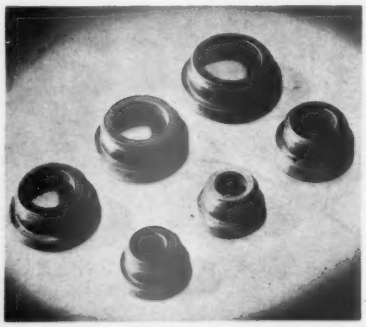
Carbon ring glands for preventing leakage on steam turbines, air and gas blowers and fans have been in wide use for over forty years. In this application the special lubricating properties of carbon in raised temperatures are the essential characteristics required. Two main factors affect the design of these glands, *i.e.*, for ease of fitting and replacement without removing the shaft or bearing they are made in segments, and secondly the high rubbing speeds, up to 10,000 f.p.m., when excessive pressures must not be exerted on the shaft. Where carbon glands are used in modern turbines they usually take the form of simple butt jointed segments surrounded by a suitable garter spring. Allowance is made for the differential expansion of carbon and steel and the joints are machined accordingly. After a short running time and under the steam pressure the joints mate closely, after

Morganite sealing rings for fluid pumps.

which there is practically no pressure on the shaft although an intimate contact is preserved. This condition makes a much tighter running joint than is possible with metal labyrinths with attendant advantages in design space. Pressures of several hundred lb. per sq. in. are sealed by means of these carbon glands.

Where operating speeds are lower, specially designed joints to eliminate axial leakage are made, and sealing pressure is allowed to be exerted by the gas or fluid medium to be sealed. In a later design of shaft gland the ring is made of unequal segments, which by a simple wedge action permits some measure of compensation of shaft sealing pressures. Radial leakage in all these shaft type glands is prevented by the side surface mating with a suitable face on the housing.

Another design of labyrinth type carbon gland is of interest, being of particular significance with the advent of the gas-turbine. It was evolved to eliminate the major proportion of leakage to which metal labyrinths are subject. The gland housing is the stator, which is provided with grooves in which the carbon segments or rings are fitted, these being of suitable section to be securely held without the aid of an encircling spring. The rotor carries fin-like annular projections positioned immediately below the bore of the carbon rings. Leakage is thus reduced since a much smaller running clearance is permissible than with metal labyrinths. Further, as the shaft whips when passing through criticals, the fins easily cut into the relatively soft carbon without damage. This is an example of the production of a special grade for a specific application, since the carbon must be soft enough to facilitate "cut in" by the fin and yet robust enough to withstand the action of the medium being sealed. Field experience with this arrangement has established that the total air leakage is reduced to as little as one-tenth of that met with in the conventional metal to metal labyrinth, the differential pressure being of the order of 50 lb. per sq. in.



Bonded rubber-carbon unit seals.

Other Applications

In addition to the above specific applications in the automotive field, carbon, metal-carbon composites, and sintered metals of specially developed compositions are extensively employed elsewhere. Some of these applications are listed below :—

(a) Carbon piston rings and rod packings for oil free compressors.
(b) Carbon gland rings for gas, steam and water turbines.
(c) Carbon valves for fluid meters.
(d) Carbon and metal-carbon vanes for rotary fluid pumps and air compressors.
(e) Large carbon and metal-carbon sealing rings for industrial pumps.
(f) Carbon arc welding rods for light gauge sheet metal.
(g) Carbon boats and tubes for melting and sintering H.M.P. metals, *e.g.*, tungsten carbide.
(h) Sintered metal friction facings for aircraft and tank starter clutches.
(i) Carbon anodes for mercury arc rectifier and transmitter valves.

Supplementary Reserve

THE War Office have announced the formation of a Supplementary Reserve. The object is to provide immediately on mobilization the technical and administrative units needed to support the fighting formation of the Regular and Territorial armies. The reserve units will consist of volunteers to provide the senior officers and N.C.Os., and will be completed ultimately with National Service men during their part-time service. Volunteers will normally be technicians whose military employment will be almost identical with their civilian occupation, and their only training liability is an annual fifteen day camp. (1946)

BRAKE TORQUE

Some Notes on the Technique of Evaluation and Measurement

By G. A. G. Fazekas, Dipl. Ing. (Zurich), A.M.I.Mech.E.

The first section of this paper explains the limitation of laboratory tests and how they should be correlated with road conditions. Test measurements and objectives are then elaborated upon, and methods of brake actuation are described. The methods of presentation, and interpretation of measurements are discussed in detail. Finally there are two Appendices, the first of which shows how the response of a brake system is calculated, and the second shows how the average torque output with regard to time differs from that referred to distance.

THE torque output of a brake invariably fluctuates even if the nominal actuation load is constant. For this reason practically all tests aim at determining the average torque output, i.e. deceleration per "stop" during which it is usual for a known amount of kinetic energy to be dissipated. Neither the method nor the interpretation of such tests is as yet standardized, and it is proposed to consider some of the fundamentals.

The subject matter centres on two main points: (i) method of brake actuation and (ii) interpretation of test results, which are preceded by a discussion of the physical background and fundamentals of test methods. Although the whole subject is treated in relation to motor vehicle brakes, the conclusions reached—qualitatively at least—are also applicable to aircraft or industrial brakes. In particular, it will be shown that some prevalent methods of measurement are subject to errors in the order of 10 to 20 per cent which do not always cancel.

Physical background

When a driver decides to brake, a certain time interval will elapse before his foot reaches the pedal. This time lag is due to the finite speed of the neuromuscular mechanism, and amounts to about 0·54 sec. with a standard deviation of 0·16 sec. (Langer 1937). Next, a certain pressure is exerted on the pedal, about which there do not seem to be any available statistical data. The writer hazards the guess that except for light check braking, pressure exerted by the foot, or manually, has probably an asymptotic character as indicated in heavy line on Fig. 1. A step function, shown dotted, may be therefore a reasonable and convenient approximation.

There are plenty of data, however, to show that once the foot has touched the pedal, subsequent deceleration of the vehicle is roughly as on Fig. 2 (Stead 1948). The deceleration vs. time function can then be divided into three parts: (a) time lag until the shoes contact the drum, (b) the "build-up" of torque, and (c) fluctuation of the built-up torque, better known as fade and grab. With a well designed mechanical brake actuating hook up (a) and (b) are usually small, although the overall mechanical efficiency may not exceed 75 per cent. With hydraulically, or servo actuated brakes one often meets the opposite. Static mechanical efficiency is very high, sometimes over 90 per cent, but there is an appreciable response lag caused by throttling. This will of course lower the average torque output per "stop". It also explains why a well designed all-mechanical hook up

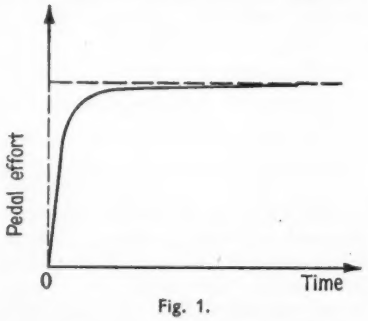


Fig. 1.

is in practice often better than a hydraulic one.

Primary factors. It would thus follow that if road and laboratory tests are to be comparable, the brakes should operate under similar loads, and should be actuated in a similar manner. Therefore the nature of the actuating force, and the static plus dynamic response of the hook up must be the same. Much of what follows will deal with these two points, but

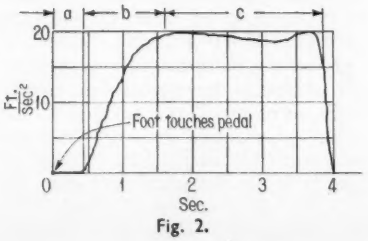


Fig. 2.

kinetic energy input, frequency of application, rubbing speed, and cooling are mentioned as being of equal importance.

Secondary factors. Even if these conditions are fulfilled, there are a number of others any of which can at times assume undue proportions. Of these the effect of rolling, frictional, and aerodynamic resistance are well known. They do not always counterbalance the rotary inertia of wheels and gears, which is rarely taken into account, although on racing or sports cars they may amount to 20 per cent of the total.

Of far greater importance is the effect of tyre slip and wheel adhesion, both causing a serious but only apparent loss of brake output, which is rarely recognized. For even assuming perfect road adhesion, it is physically impossible to transmit a force through a moving tyre without some slip (Olley 1946). The angular movement of the wheels is thus bound to be less than would correspond to (stopping distance)/("effective" tyre radius). Therefore the brakes are effective only part of the way, which is equivalent to reducing the drum radius in proportion to the slip. How much of the observed total slip is caused by imperfections of the suspension alone, is not known, but in the writer's experience even shock absorber setting has often a surprisingly marked effect on brake performance. This will be perhaps evident when considering that the total slip of a wheel is in the order of 20 per cent for maximum road grip (Klaue 1939) and its value is about 10 per cent at 20 ft/sec.^{2} retardation. Furthermore, since wheel slip is a function of the retarding force, it is in general different on the front and rear wheels, depending on braking ratio, weight transfer, suspension, condition and pressure of tyres, etc. Consequently, maximum retardation of the vehicle does not in general coincide with the maximum grip of all its wheels.

It would thus seem that within about ± 10 per cent brake performance can be affected by the "road-holding" of the vehicle. This, incidentally, may explain the well-known discrepancies in the measurement of road friction.

Response lag. Mention has been made of the delayed response of hydraulic, or like actuating systems. For laboratory tests an equivalent system should

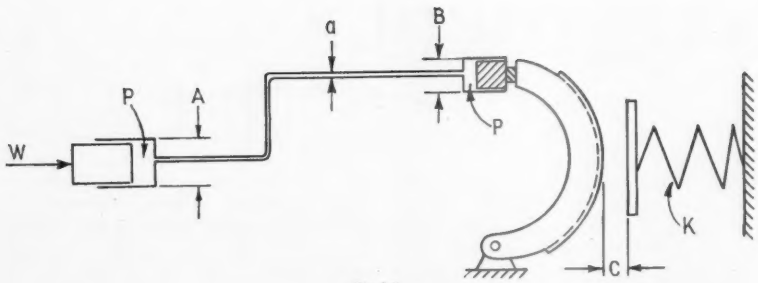


Fig. 3.

be rigged up. To show the order of magnitude of this point, the response of a perhaps fairly representative system is calculated through in Appendix I, together with the derivation of general formulae assuming zero static friction. The usual 10 to 20 per cent static friction loss can be, however, algebraically deducted.*

The example selected refers to a pair of 12 in. × 1¾ in. two-leading-shoe brakes with 1⅛ in. bore wheel cylinders and 0·045 in. shoe-drum clearance. The hook up comprises a 1 in. bore master pump, 8 ft. 0 in. of $\frac{5}{32}$ in. bore piping, and two flexible hoses at 14 in. long by ⅛ in. bore. Half of the main piping is shared by both brakes, and the remaining half is divided into two equal branches at 24 in. long. The equivalent hydraulic system is shown diagrammatically on Fig. 3 and has the following main data :

A = 0·7850 in.²
B = 0·9940 in.²
a = 0·0192 in.²
L = 9 ft. 3 in.
C = 0·360 in.
K = 2,600 lb/in.
W = 393 lb.
= 500 × 0.785

The response of this system to a step function of amplitude (W) acting on (A) is plotted on Fig. 4 for zero static friction and for a usual brake fluid of 87.7 CS viscosity at 65 deg F.

Let it be supposed now that the vehicle is to be stopped from 60 m.p.h. with a maximum retardation of 15 ft/sec². The above value of (W) corresponds roughly to this condition. Ideally, the stopping distance would be 258 ft., but it will be affected by varying fluid temperature as shown in Table I.

If it is further supposed that there is say 10 per cent static friction in the hook up, the same output would be obtained by increasing (W) from 393 to 432 lb. The corresponding ideal stopping distance would be of course 258/1·10 = 235 ft., adding 10 per cent to the above figures.

It may be of some interest to note that the 2⅛ in³ fluid content of the pipe line is equivalent to the inertia of a 112 lb. dead weight attached to master piston (A). That an equivalent hook up is essential for other than purely comparative laboratory tests, seems thus rather obvious. Also, if the piping arrangement is grossly unsymmetrical, the vehicle may "pull," or the front/rear ratio may get out of control.

*Although seal friction increases in general with velocity and pressure (White and Denny 1948), this seems to apply to large movements only. For small movements and low velocities the writer's own tests indicated a substantially constant friction, in the order of 50 psi/in² piston area.

Table I.

At a fluid temperature	25	45	65	°F
Fluid viscosity	425	183	87·7	CS
Due to contact time lag	22·4	9·4	4·2	ft.
Due to build up	15·0	6·3	3·0	ft.
Total	37·4	15·7	7·2	ft.
This amounts to	14½	6	2¾	%

Nature of brake tests

Logically, a brake has to stand up to a certain duty, i.e., to a certain frequency, quantity and rate of energy dissipation at a certain rate of cooling. Fundamentally, all brake tests seem to fall thus into three categories: (i) what is the maximum duty the brake can sustain, i.e., rating with regard to lining wear, fade, etc., (ii) what is the relationship between torque output and actuating force input, known as performance test, and (iii) how does the brake behave during its lifetime, when in "normal" use, known as bedding tests.

Rating tests. Apart from trivial mechanical problems (e.g. strength of shoes) the test objective is to find such friction properties as wear, fade, heat checks, etc., as a function of the duty imposed. Laboratory tests (Sisman 1936) seem to be based on the tacit premise that (i) a certain "duty" as defined above adequately represents the statistical average of field conditions, as does (ii) the reaction of the brake thereto, and that (iii) the brake will stand up to its task. Should it flag it is an empirical fact that the operator tends to ease the load, often subconsciously. These qualifying conditions were raised in order to explain why these essentially research tests are sometimes at variance with practical experience.

Laboratory tests entail (a) constant dynamometer speed, (b) a "time cycle" device to declutch the drive and to apply the brakes periodically, (c) a "constant torque" device (Sisman 1936) to keep the rate of energy absorption constant, this being roughly equivalent to constant stopping time, and (d) cooling arrangements to a definite pattern, the importance of which cannot be over-emphasized.

In the writer's experience, whereas rating tests are absolutely essential to find the endurance limit of brakes, unfortunately they have to be relegated to the laboratory *in toto*. For sustained tests of such nature impose so great a strain on the driver as to become unreliable when carried out in the field.

Performance tests. With the test objective of correlating torque output with actuating force input, theoretically the correct procedure would be to analyse statistically the results obtained with a fleet of vehicles, all of one type. This stipulation is included because not only the frictional but also the mechanical properties of the entire brake installation, hook up included, have to be tested. The duty imposed ought to resemble therefore the statistical averages of the entire range within which the brake is used.

If tropical and arctic tests are treated as separate items, and if the ambient temperature is reasonably constant, the "duty" is defined by three variables only, because vehicle speed, i.e., kinetic energy, automatically fixes the rate of cooling as well. The three variables are : (i) vehicle speed,

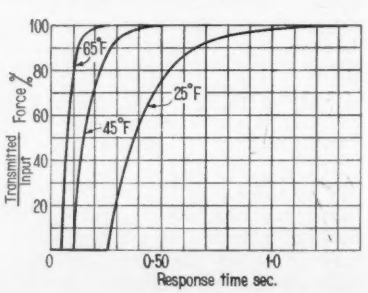


Fig. 4.

(ii) frequency of application and (iii) intensity of deceleration, the function connecting them with the actuating load may be thus defined as brake performance. For experimental purposes any two variables can be kept constant, and the behaviour of the other two studied. Consequently three types of tests can be carried out.

The usual deceleration vs. pedal load test at constant frequency of application and from constant speed (e.g. 30 m.p.h.) is one of the three possibilities. A complementary procedure, known as fade test, consists in keeping vehicle speed and pedal load constant, and measuring deceleration as a function of the number of stops. Another form of fade test is to measure the variation of pedal load that is necessary to keep deceleration constant (Sisman 1936). These three tests have their counterparts in appropriate laboratory tests. On physical grounds alone they seem equally justified. From the physiological aspect, however, the writer believes that tests on the constant load application basis are a closer approximation to what happens in practice if single stops only are to be considered, although he has no statistical data to prove it directly. For this reason and because of greater ease, the constant load application basis seems more suitable for the present purpose.

Bedding tests. In the absence of relevant statistical data the writer hazards the guess that in normal usage as the torque output varies, the driver tends to correct it, but only partially. If the brake's life history is to be studied, the test conditions should thus lie half way between rating and performance tests.

Brake drum temperature. No reference has been made so far to brake drum temperature although, owing to its great influence, it might appear essential to link it somehow with brake tests. This omission is due to the fact that temperature *per se* is only an incidental phenomenon coincident with, but not a yardstick of, brake performance. Its use as a yardstick seems rather misleading because: drum surface temperature originates from two main sources: (i) a steady level is fixed by the "duty"; (ii) a series of peaks, depending on the nature of the co-acting surfaces, are superimposed (Bowden 1949, Parker 1948). Hence during the actual process of braking, drum temperature is partly independent of the duty imposed. That is, unless the "duty" is interpreted to mean some thermometer reading and not the number and intensity of stops at certain intervals. This *a priori* tenders further problems of measurement, interpretation, etc., irrelevant except so far as cooling is concerned. In addition matrix temperature is fixed by the duty imposed, whereas the converse is not true, since a certain temperature can be reached in many ways. Considering, however, that history, better known as recovery, has a profound influence on friction (Saunders 1946) surface temperature appears to be a redundant co-ordinate. Matrix temperature is thus a significant and true measure of the rate of cooling alone.

To give an example, with a certain racing car brake the writer found the drums badly blued all over on using a hard moulded lining with a lot of metal chips in it. Exactly the same brake, under the same dynamometer duty but with a soft woven lining only straw coloured a similar drum. Metallurgical evidence indicated a temperature difference in excess of 100 deg C. The same recipe seems to have proven itself on a popular make of vehicle, which has soft linings fitted to highly rated brakes to prevent judder caused by hot spots. This substantiates the observations made in (ii) first column.

Method of brake actuation

In the absence of a suitable automatic device, in road tests, the driver's skill has to be relied upon to maintain the stipulated constant actuation load when brake performance is measured. Bedding tests on the road are done just by some hard driving. Fortunately the human element can be largely eliminated from laboratory tests for which the necessary equipment will be described below.

Constant torque device. Fundamentally, the problem of maintaining constant torque is about the same as say, governing a steam turbine. There does not seem to be any reason why the control gears should not be quite simple, provided that inertia, friction and response lag are kept low. For this reason the hook up should have a negligible response lag in the present case. There being sufficient published data on similar mechanisms, the subject will not be further treated here. (Smith 1944.)

Constant load device. As mentioned before, the aim is to apply a step function and since it is accompanied by appreciable movement, inertia effects should be negligible. (If dead weights are used, the effective load is quite unpredictable during the first half sec. or so, just when the brakes do most work.)

Fig. 5 shows diagrammatically a compressed air device which can be coupled up to the ordinary compressed air main available in workshops. It comprises a pressure reducing valve, an air reservoir, a quick-opening valve and a piston which actuates such hook up as exists. Some laboratories the writer has been in contact with now use this arrangement.

Fig. 6 shows as an alternative a purely mechanical device, made up from a linear response toggle spring and a preloaded compression spring both having the same rate (k). If now the toggle spring halves are considered as a single unit, its free length corresponding to the horizontal position of the toggle arms, with the notation as on Fig. 6, the load (Q) in the toggle is

$$Q = k.y.\operatorname{cosec}\phi \qquad (1)$$

With x_o = the equivalent free length of compression spring, its thrust (P) is

$$P = k.(x_o - 2x) \qquad (2)$$

On compounding the forces horizontally, the resultant (W) is

$$W = P + 2Q.\cos\phi = k.x_o + 2k.(y.\cot\phi - x) = k.x_o = \text{constant} \qquad (3)$$

One end of the compression spring is threaded into the helical grooves

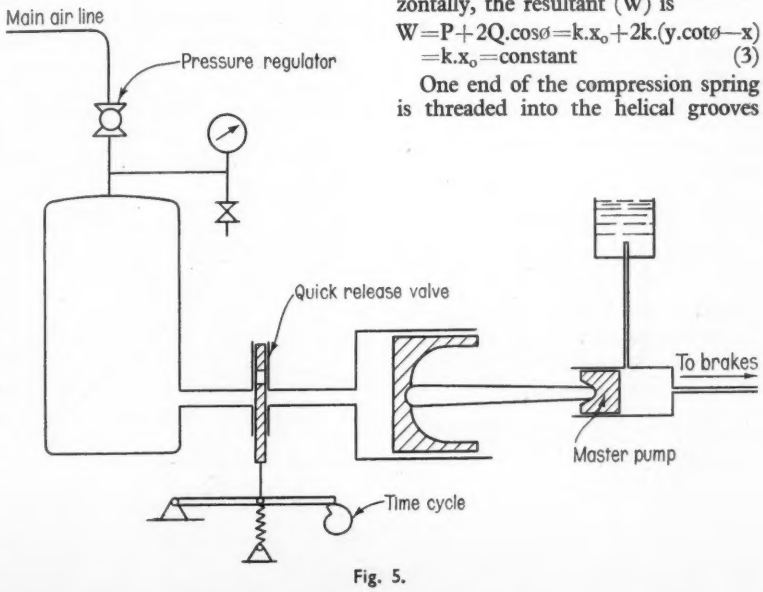


Fig. 5.

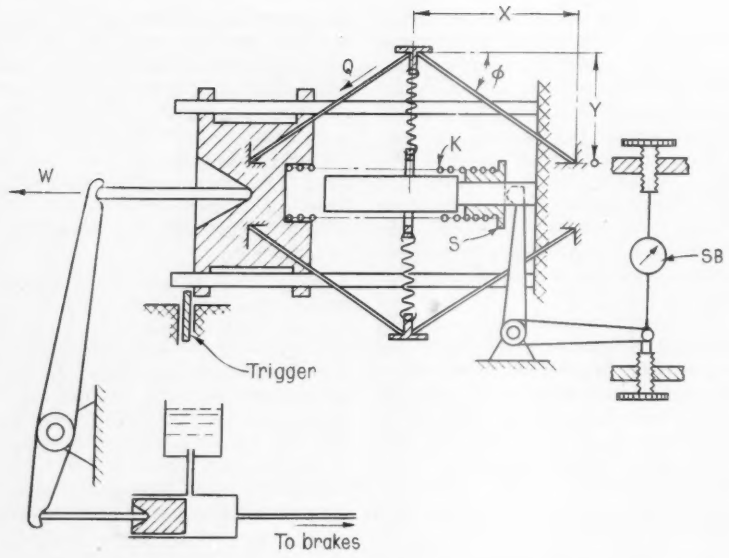


Fig. 6.

of sleeve (S) which can be turned to match spring rates. The position of the same sleeve (S) is adjusted through a lever arrangement and the load read off directly from spring balance (SB). With this device only about half of the total inertia is effective, amounting to a few pounds for some 200 lb. thrust.

Bedding device. In between performance tests, fade tests, etc., hundreds of stops are carried out on dynamometers. As already mentioned, during these stops the desideratum is to compensate fade (or grab) partially. This can be done rather easily with the negative feed-back device shown in its simplest form on Fig. 7.

The torque arm is loaded by a force (W) and at (N) it actuates a master pump with plunger area (A) connected to wheel cylinder(s) of area (B). For the direction of rotation shown torque (T) opposes (W) with the result that this device transmutes a leading into a trailing, and a trailing into a super trailing shoe brake. It can be adjusted to cut down fade or grab to any desired degree.

For if (K) is the torque factor and (R) the radius of the brake

$$T=K.R.\left(\frac{W.L-T}{N}\right)\frac{B}{A} \qquad (4)$$

Let it be supposed now that the friction drops slightly, as a result of which the torque factor diminishes by (dK) and the torque by (dT). Then (dT/T) will be a measure of fade (Fazekas 1951). On rearranging the above equation and differentiating it

$$\frac{dT}{T}=\frac{dK}{K}\left[1+\frac{K.R.B.}{A.N}\right]^{-1} \qquad (5)$$

whereas without feed back the expression in the parentheses=1. With feed back, if say the torque fluctuation is to be reduced to ¼ of its normal value, the actuating load (W) has to be increased four times. Because if the expression in the parentheses of equ. 5 be denoted by (G), equ. 4 can be rewritten as

$$T=(W/G).(K.R.L.B/N.A) \qquad (6)$$

from which it follows that for a given (T), (W/G) must be constant.

A small dashpot, indicated on Fig. 7, the writer found useful, but not always necessary to steady the apparatus.

Method and interpretation of measurement

In most industrial applications, and certainly on motor vehicles, stopping distance is the primary criterion of brake performance. The torque output should be thus averaged with regard to distance and not time. That the two averages are not the same, and in fact differ appreciably, is known (Langer 1937). It is also shown in Appendix II that if (T) be the torque at time (t), (T_t) its time and (T_s) its distance average, with (t_o) as the stopping time

$$\frac{T_t}{T_s}=2.\left[1-\frac{\int_o^{t_o}\int_o^{t} T.dt^2}{T_t.t_o^2}\right] \qquad (7)$$

and in general the expression in the parentheses is not equal to ½.

To illustrate the significance of the above, let it be supposed that the torque vs. time function be given by

$$T=T_o.(1-\exp^{-bt}) \qquad (8)$$

which about represents the response lag function deduced in Appendix I and approximates test results. Let it be further supposed that the damping (b) is to be varied and it is required to find its effect on (T_t) and (T_s). The nature of equ. 8 is shown on Fig. 8 and since the slope at O is (T_o.b) an easly visualized parameter would be (α), as indicated. Between the two extremes shown, the value of (α) lies between zero and 1. For various values within this range the

$$\text{Braking efficiency}=\frac{\text{Average torque}}{\text{Maximum torque}} \qquad (9)$$

is plotted on Fig. 9 both for time (T_t/T_m) and for distance based (T_s/T_m) averages. It will be seen that for small values of (α) the loss on distance basis is about twice that on time basis.

It would thus seem that quantitative assessment of brake performance on anything but distance basis can be rather deceptive.

Road tests. To obtain the distance based average retardation, one can use either the conventional fifth wheel method, or a continuously recording decelerometer and double integration (Gough 1949). Of the two the fifth wheel method has the advantage of directly estimating vehicle speed and hence the kinetic energy absorption. Its further advantage is the direct and quick evaluation of test results which seems to more than save the extra time necessary to fix it up. Also if a fifth wheel is used in conjunction with a Tapley Meter set for "maximum," it allows a quick direct measurement of Braking Efficiency as well. Perhaps the ideal arrangement would be to

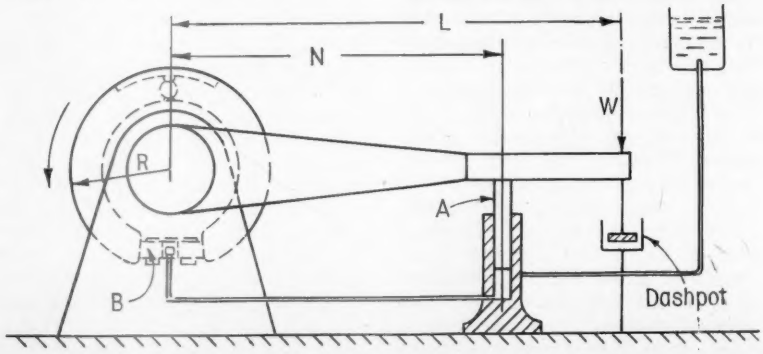


Fig. 7.

provide a continuously recording decelerometer with a distance base derived from a fifth wheel. (Electronic instruments, in the writer's opinion, are not sufficiently robust for the present purpose.)

The rather conventional U tube decelerometers tend to register the maximum rather than the average retardation and over-estimate brake performance even on a time basis. The magnitude of the error is in the order of 20 per cent (Langer 1937). In fact often more reliable results can be obtained by using a stopwatch and the vehicle's speedometer.

Little attention has been devoted so far to the measurement of pedal load and to ensuring that it should conform to a definite pattern. In the writer's opinion, this is one of the main outstanding problems in brake testing technique. It is even more important than providing a continuously recording decelerometer, or the like with a pressure recording stylus (Kent 1946).

The retardation vs. pedal load characteristic obtained by any of the foregoing methods does not of course provide a true picture of brake performance *per se*, for reasons mentioned on p. 2 *ante*. To obtain a complete picture it is necessary to know (i) front wheel slip, (ii) rear wheel slip, (iii) rotary inertia of wheels and gears and (iv) aerodynamic + frictional resistances, all of which are usually neglected. On passenger vehicles (iii) and (iv) usually balance each other, but because of (i) and (ii) brake performance is underestimated. The error may be in the order of 5 to 30 per cent. It is thus advisable to fit a revolution counter to at least one wheel; incidentally, the slip values obtained may be regarded as a good comparative measure of the vehicle's road holding ability.

Dynamometer tests. In laboratory tests measurement of the actuating load presents no difficulties, a pressure gauge or a spring balance is quite satisfactory. The correct quantitative interpretation of constant torque tests can be carried out however only if the load fluctuation is recorded on a distance basis. The recording gear should be coupled therefore to the flywheel shaft.

Similarly, the average torque per stop should be evaluated on a distance basis. The writer found that perhaps the simplest way to do it is to couple an ordinary revolution counter to the flywheel through a (small friction) clutch and a (helical) torsion spring, to cushion the shock. It is rather important though to couple the triggering-off mechanism of the clutch to the master pump end of the (equivalent) hook up in order to include fully the pump's response lag. This arrangement is very accurate and is better than that of the conventional stopping time measuring device. The reason is because a stopwatch has to be started and stopped, and both processes involve some errors, whereas the revolution counter has to be only started.

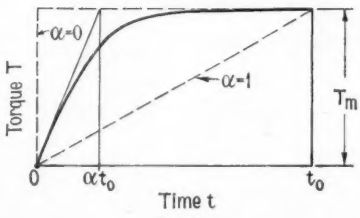


Fig. 8.

Apart from the mean torque per stop it is rather useful to study in detail the torque fluctuations throughout the stop. Perhaps the most appropriate instrument to do it with is a converted mechanical engine indicator. Its accuracy satisfies the most exacting requirements since the time period of the recording is from 2 to 10 secs., and the torque arm movement can easily be kept to fractions of an inch to avoid inertia effects.

Summary and conclusions

In the foregoing an attempt has been made to clarify what constitutes brake performance, how it should be measured and why. Only the basic principles of the most important test equipment and measuring methods were described. These centred around the following findings:

(a) All test results and conditions—with the one exception of rate of cooling—should be specified and expressed respectively in terms of lb., ft., and sec., without introducing "drum temperature".
(b) Brake performance should be measured on the "constant application load" basis, as distinct from
(c) rating the frictional properties for which the "constant torque" type of dynamometer tests seem correct.

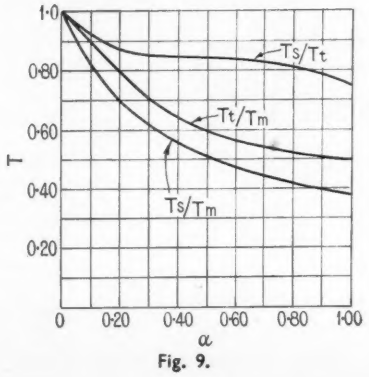


Fig. 9.

(d) Torque performance should be averaged on distance and not on time basis because
(e) time basis underestimates as a rule and
(f) the frequently used U tube decelerometers ignore altogether losses caused by the response lag of the hook up. For this reason and for other than purely comparative tests.
(g) the hook up employed on dynamometer tests should be statically and dynamically equivalent to the one used on the vehicle, whilst the
(h) brake actuating mechanism employed for dynamometer tests should have negligible inertia. For close agreement between road and dynamometer tests it is further essential to measure the wheel slip, whilst rotary inertia of wheels and gears may be, but is not always, counterbalanced by the aerodynamic + frictional resistance of the vehicle.

By observing these points, road and dynamometer tests appear to be so closely correlated that the difference between the two need be caused only by the test driver's skill. At first it may seem that road tests could be avoided altogether, but unfortunately this is not so, because as a prerequisite of accurate dynamometer tests it is essential to know (*a*) wheel slip and (b) cooling, neither of which can be reliably predicted without some preliminary test runs on the vehicle in question, nor is it possible to avoid a check (c) for trivial mechanical points to see whether the "vehicle suits the brakes", and finally very little is as yet definitely known about (d) the "feel" of brakes (Fazekas 1950). On the other hand, once these points are established, much of the laborious road testing for wear, life, recovery, etc., can be relegated to the laboratory, and it is hoped, with greater measure of safety than was possible hitherto.

Finally, the writer wishes to acknowledge his indebtedness to the Directors of Girling Ltd., for facilities granted which led to the compilation of these notes.

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APPENDIX I

Fig. 3 shows diagrammatically a simplified hydraulic brake actuating system in which viscous friction is assumed to be the only resistance. It is further supposed that a single spring of constant rate (K) represents all elastic deformations, i.e., the pipe line and the fluid. The hydraulic cylinders are assumed to be infinitely rigid. Only the equivalent inertia of the fluid in the pipes is taken into account, against which the inertia of pistons, shoes, and drum are negligible. The fluid flow is assumed to be isothermal and laminar, with negligible exit and entry losses, and not materially influenced by the pressure viscosity effect.

The treatment is divided into four parts: the first one deals with the time lag required to take up a certain shoe-drum clearance, and will be referred to as contact lag. The second part shows the force build-up, the third deals with the effects of the first two on brake performance, and the fourth part deals with the application to a practical example.

1. CONTACT LAG.

When the motion is referred to the master piston of area (A) with movement (y) Fig. 3, the equivalent mass (m) of the fluid in the pipe of area (a) and length (L) is

$$m=\rho.L.a.(A/a)^2 \qquad (1.1)$$

where ρ = density.

From Poiseuille's law it follows further that the rate of fluid flow is, with (B) as the slave piston area and (x) its movement

$$B.\dot{x}=(P-p).a^2/8.\pi.L.z \qquad (1.2)$$

where z = absolute
" p = pressure in B
" P = " " A

Viscosity: The equation of continuity results in

$$y.A=B.x\ ;\ \dot{y}.A=\dot{x}.B\ ;\ \ddot{y}.A=\ddot{x}.B \qquad (1.3)$$

Since the slave piston (B) is not loaded, p = O and the equation of motion is for a step function of amplitude (W)

$$\ddot{y}.\rho La.(A/a)^2+\dot{y}.8\pi zL.(A/a)^2=W \qquad (1.4)$$

Writing now

$$q=W/\rho La(A/a)^2 \text{ and } c=8\pi z/\rho a \qquad (1.5)$$

equ. 1.4 reduces to

$$\ddot{y}+\dot{y}.c=q \qquad (1.6)$$

the solution of which is

$$y=v_o.t.\left[1-\frac{1-\text{exp.}-ct}{ct}\right] \qquad (1.7)$$

where

$$v_o=q/c=W.(a/A)^2/8\pi zL \qquad (1.8)$$

= terminal velocity

For ct > 5 equ. 1.8 can be well approximated by

$$y\sim v_o.t \qquad (1.9)$$

and hence the time required to cover a clearance (C) at (B) is

$$t_c\sim C\frac{8\pi zL}{W}\left(\frac{A}{a}\right)^2.\left(\frac{B}{A}\right) \qquad (1.10)$$

2. FORCE TRANSMISSION.

When pressure (p) acts on (B) we have from equ. 1.2

$$P=p+8\pi LzA\dot{y}/a^2 \qquad (2.1)$$

The transmitted force is then

$$p.B=K.x=K.y.(A/B) \qquad (2.2)$$

The equation of motion is thus for (A) and for a step function of amplitude (W)

$$\ddot{y}.\rho La.(A/a)^2+\dot{y}.8\pi zL.(A/a)^2+K.y.(A/B)^2=W \qquad (2.3)$$

On writing now

$$n^2=K.(a/B)^2/\rho La \qquad (2.4)$$

and with reference to equs. 1.5 we have

$$\ddot{y}+\dot{y}.c+y.n^2=q \qquad (2.5)$$

the solution of which is

$$y=s+C'.e^{r't}+C''.e^{r''t} \qquad (2.6)$$

where

$$s=q/n^2=(W/K).(B/A)^2 \qquad (2.7)$$

= full static deflection

Since with a sluggish brake, which is the one of interest, $(n/c)<1$, only the aperiodic solution will be considered, when the characteristic roots are

$$r'=\tfrac{1}{2}.\left[-c+c.\sqrt{1-4n^2/c^2}\right]\sim-n^2/c$$
$$r''=\tfrac{1}{2}.\left[-c-c.\sqrt{1-4n^2/c^2}\right] \qquad (2.8)$$
$$\sim-c+n^2/c\sim-c$$

The integration constants (C') and (C'') are from the boundary conditions y = O, $\dot{y}=v_o$ at t = O

$$C'=\frac{V_o+r''s}{r'-r''}\sim\frac{v_o-c.s}{c-n^2/c}\sim v_o/c-s \qquad (2.9)$$
$$C''=-C'-s\sim-v_o/c$$

The complete solution is thus approximately

$$y\sim s+(v_o/c-s).\text{exp.}-n^2t/c-(v_o/c).\text{exp.}-ct \qquad (2.10)$$

In this the last term being negligible for t » O, and on writing

$$b=n^2/c=\frac{K.(a/B)^2}{8\pi Lz} \qquad (2.11)$$

we have

$$y=s.(1-e^{-bt})+\frac{v_o}{c}e^{-bt} \qquad (2.12)$$

Now the force transmitted is

$$K.x=K.y.(A/B) \qquad (2.13)$$

whereas without response lag it would have been W.B/A. The kinetic efficiency (E) is thus at the instant (t)

$$E=K.y.(A/B)^2/W=y/s$$
$$=1-e^{-bt}+\frac{v_o}{cs}e^{-bt} \qquad (2.14)$$

3. BRAKING EFFICIENCY.

Let it be supposed that a vehicle of mass (M) travelling with velocity (V) is to be arrested by a force (F), as given by

$$F=F_o.(1-e^{-bt}) \qquad (3.1)$$

which is a close approximation of equ. 2.14 for large values of (c). The stopping time (t_o) can be then obtained from the first integration of motion which results in

$$M.V=F_o\int_0^{t_o}(1-e^{-bt}).dt\sim F_o.(t_o-1/b) \qquad (3.2)$$

for $bt_o>5$ which is the case in practice. Hence with (t*) as the stopping time in the ideal case of $b=\infty$, i.e. c = O,

$$t_o=\frac{M.V}{F_o}+\frac{1}{b}=t^*+\frac{1}{b} \qquad (3.3)$$

The stopping distance (S) can be obtained from a repeated integration of equ. 3.2 and results, with a contact lag (t_c) as given by equ. 1.10,

$$S=V.(t_o+t_c)-\frac{F_o}{M}\int_O^{t_o}\int_O^{t}(1-e^{-bt}).dt^2 \qquad (3.4)$$
$$\sim V.(t_o+t_c)-\frac{F_ot^2}{2M}\left[1+\frac{2}{(bt_o)^2}-\frac{2}{bt_o}\right]$$

if $bt_o>5$, which is usually the case. In the ideal case of $b=\infty$ the stopping distance would have been

$$S_o=V.t^*-\frac{F_o.t^{*2}}{2.M}=\tfrac{1}{2}V/f \qquad (3.5)$$

where f = deceleration. On combining equs. 3.4 and 3.5 and after some manipulations we have

$$S=S_o+V.(t_c+1/b)-F_o/2Mb^2 \qquad (3.6)$$

Since

$$f=V/t^*=F_o/M \qquad (3.7)$$

the ideal stopping distance is increased by

$$\Delta S=V.(t_c+1/b)-\tfrac{1}{2}f/b^2 \qquad (3.8)$$

the relative magnitude of which is

$$\frac{\Delta S}{S}=\frac{2.f.(t_c+1/b)}{V}=\frac{2.(t_c+1/b)}{t^*} \qquad (3.9)$$

NB. On a time basis the loss would have been $(t_c+1/b)/t^*$ which is half of the above value.

4. EXAMPLE.

For a pair of 12" × 1¾" two leading shoe brakes with 1⅛" bore wheel cylinders the data are

a. Displacement at 1000 psi. pressure from experiments

(elastic) wheel cyl. fluid displacement/brake	0·141in³
hose expansion/brake	0·035 "
compression of 3 in³ brake fluid	0·016 "
Total displacement/brake	0·192 in³
equivalent piston movement for both brakes	0·384 "

Shoe-drum clearance/shoe	0·045"
Total piston movement due to clearance	8x 0.045 = 360"

Therefore K = 2600 lb/in & C = ·360"

b. Piping. Usual installation length = 8′ 0", bore = $\frac{5}{32}$" of which half is common to both brakes and half is branched. Two hoses of ⅛" bore 14" length have to be added to the above.

The equivalent lengths are thus

Common pipe	48 in.
2 branches at 24" with half flow	24 "
2 hoses at 14" with half flow $14\times(5/4)^4=$	34 "
Allowance for corners, bends, etc.	5 "
Total equivalent length	111 in.

Therefore L = 9′ 3"

c. Fluid data. referring to a normal brake fluid

Temperature °F	25	45	65
Viscosity Poises	4·08	1·70	0·79
Viscosity 10^6xReyns	59·1	24·7	11·5
Specific gravity	0·96	0·93	0·90

d. Main data are thus with reference to Fig. 3 and for 500 psi P.L.P.

W = 393 lb.	a = 0·0192 in²	L = 111 in.
K = 2600 lb./in.	A = 0·785 "	A/a = 41
C = 0·360 in.	B = 0·994 "	B/A = 1·26

TEMPERATURE	°F	25	45	65
m equ.1.1	lb.	125	121	117
c equ.1.5	sec^{-1}	855	368	177
q equ.1.5	in/sec^2	1210	1250	1343
v_o equ.1.8	in/sec	1·415	3·400	7·570
t_c equ.1.10	sec	0·254	0·106	0·048
n^2 equ.2.4	sec^{-2}	5020	5190	5365
s equ.2.7	in.	0·241	0·241	0·241
b equ.2.11	sec^{-1}	5·87	14·10	30·40
t_c+1/b	sec	0·424	0·177	0·081
$t^*=60\times1.47/15=$	sec		5·88	
$\frac{1}{2}f/b^2$ for f=15 ft/sec²	ft.	0·216	0·038	0·008
ΔS equ.3.8	ft.	37·40	15·60	7·12
$2.(t_c+1/b)/t^*$	0	0·145	0·060	0·0275

APPENDIX II

Let it be supposed that a variable torque (T) arrests a flywheel of inertia (I) from an initial angular velocity (V). The equation of motion leads then to

$$I.V=\int_0^{t_o} T.dt \qquad (4.1)$$

where t_o=stopping time.

The time averaged mean torque (T_t) is then

$$T_t=\frac{1}{t_o}\int_0^{t_o} T.dt \qquad (4.2)$$

If however the torque be averaged with regard to (angular) movement, as per definition it will be

$$T_s=I.V^2/2.S \qquad (4.3)$$

where S=(angular) movement to standstill, which can be obtained from a second integration of equ.1.

$$S=V.t_o-\frac{1}{I}\int_0^{t_o}\int_0^{t} T.dt^2 \qquad (4.4)$$

Substituting now for (S) in equ. 4.3 and rearranging we have

$$T_s=\frac{I.V^2}{2.V.t_o-\frac{2}{I}\int_0^{t_o}\int_0^{t} t.dt^2} \qquad (4.5)$$

which combined with equs. 4.2 and 4.1 results in

$$\frac{T_t}{T_s}=2.\left[1-\frac{\int_0^{t_o}\int_0^{t} T.dt^2}{I.t.t_o^2}\right] \qquad (4.6)$$

Let it be supposed now that the torque vs. time function is given by

$$T=T_o.(1-e^{-bt}) \qquad (4.7)$$

Then with reference to Fig. 8 on writing

$$u=b.t_o \qquad (4.8)$$

$$a=(1-e^{-u})/u \qquad (4.9)$$

because

$$T_o b.\ t_o=T_m=T_o(1-e^{-u}) \qquad (4.10)$$

With reference to equ. 4.2

$$T_t=\frac{T_o}{t_o}\int_0^{t_o}(1-e^{-bt}).dt$$

$$=T_o.\left(1-\frac{1}{u}+\frac{e^{-u}}{u}\right)$$

On dividing it by T_m

$$\frac{T_t}{T_m}=1-\frac{1}{u}+\frac{1}{e^u-1} \qquad (4.12)$$

Quite similarly

$$\frac{T_s}{T_m}=\frac{(u-1+e^{-u})^2}{[u^2+2.(1+u).e^{-u}-2].(1-e^{-u})} \qquad (4.13)$$

When $u=\infty$ tends to zero and equs. 4.9, 4.12 and 4.13 reduce to

$$a=1/u \qquad (4.14)$$

so that

$$T_t/T_m=1-1/u=1-a \qquad (4.15)$$

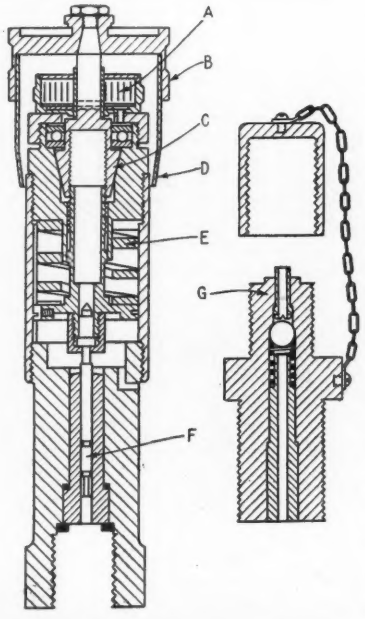
and

$$\frac{T_s}{T_m}=\frac{(1-u)^2}{u^2-2}=\frac{(1-a)^2}{1-2a^2} \qquad (4.16)$$

A NEW PEAK PRESSURE INDICATOR

A DESCRIPTION was given in the February 1951 issue of *The Automobile Engineer* of a peak pressure indicator introduced by Dobbie McInnes Limited, Broomloan Road, Glasgow, S.W.1, which they are making under Sulzer patents. It is designed for engine cylinder pressures up to 1,500 lb. per sq. in. The firm has now produced a further model for indicating maximum pressures up to 15,000 lb. per sq. in., and it has a special use for oil engine fuel systems. It is of particular value in providing early warning of choking atomizers which, especially with the high viscosity oil fuels now being used on an increasing scale, may produce undesirable fuel line pressures 100 per cent and more above normal. To reduce the cost of an outfit for a multi-cylinder engine, and to avoid the need for acquiring one indicator for every cylinder, a simple automatic cock has been designed. One of these is connected to each fuel line, and one or two indicators only can be rapidly transferred from cock to cock.

Referring to the drawing the operation is as follows:— a highly finished plunger F loaded by a calibrated spring E rises under the influence of the fuel pressure, and lifts a conical nut C off its seat. During the first cycle, a clock spring A returns the nut downwards to the seat, and the rotation of the nut is indicated by a graduated scale D on a thimble B which turns with it. During the second cycle the plunger, which has been locked in the "up" position by the nut, rises a little further and is again locked by the nut. This is repeated over a few cycles until the thimble stops rotating and the maximum pressure in the fuel line is indicated on the scale. For a repeat check, the thimble is rotated by hand a quarter turn and released. The thimble is graduated from zero to 10,000 lb. per sq. in. by 100 lb. per sq. in. divisions, and it will make another half turn for 15,000 lb. per sq. in. in all. The indicator may be calibrated on an ordinary high pressure gauge tester.



Dobbie McInnes peak pressure indicator.

The automatic cock G is normally screwed to a T-piece in the fuel line and has a tail ¾ in. B.S.P. × 1½ in. male thread or otherwise as required. It automatically opens when the indicator is mounted, and shuts when the indicator is removed. As there are no handles to manipulate, indicating a line of cylinders is a very quick process.

After bench tests at the factory, the indicator and cock were tried out at various speeds and pressures at the works of well-known manufacturers of fuel injection equipments. Engine tests were conducted during the shop trials of a diesel auxiliary engine running under a main engine at sea. During these last tests not only were previous figures conistently confirmed while nearly fifty readings were taken, but the indicator proved itself entirely suitable for ships' engine-room usage. (1954)

OIL FILTERS

By-pass v. Full Flow Tests Carried out by Vokes Ltd.

THE relative merits of by-pass, and full flow oil filtration systems for automobile engines have long been the subject of controversy. It is also a subject which has become of increasing importance in recent years with the expansion of our export markets in countries where dusty road surfaces are more prevalent than hard ones. A glance through recent engine specifications will show that opinions are fairly evenly divided on this subject. Some manufacturers install one system exclusively, whilst others use both with a tendency towards the full flow type on larger vehicles. Indeed, a very small minority use no filter at all, and it is rather surprising to note that on several popular American cars the by-pass filter is an optional extra.

A report issued by Vokes Ltd., of Guildford compares engine wear occurring with a Vokes full flow filter fitted with that obtained when a filter of the partial flow type is incorporated, and it would appear to add some useful facts to the somewhat limited data published so far on this subject. The report is a sequel to a similar one reviewed in the issue of *Automobile Engineer* for January, 1948. This report compared engine wear without filters with that obtained with a Vokes air filter alone, and then with both the Vokes air, and full flow filters fitted.

As is generally known, in the full flow system all the oil circulating is passed through a filter, whereas in the partial flow system only a portion of the oil is by-passed from the main flow through a filter. Protagonists of the by-pass scheme claim that it is better to remove a high proportion of the finer particles of foreign matter from the oil by passing a part of the circulation through a fine filter, than to remove only the coarser particles which is all that is possible with a similar size of full flow filter. It is also claimed that the partial flow filter removes the foreign matter from the total supply of oil at a faster rate than that at which it enters, thereby keeping the total supply of oil in the sump in a much cleaner state generally than is the case when the full flow filter is fitted.

The engine used for the tests was a four-cylinder 8 h.p. four-stroke petrol engine manufactured by the Standard Motor Company. As far as the test was concerned the most important engine details were :

57 mm. bore, 100 mm. stroke
Side-by-side valves
Three-bearing crankshaft
Removable main and big end shells
Chromium iron cylinder block
Downdraught carburettor

Lubrication was by gear type pump supplying oil at a working pressure of 45 lb. per sq. in. to the main bearings and camshaft gearing. Splash feed lubrication was supplied to the little end bearings and valve mechanism.

Engine cooling was effected by an external pump which drew water from a cooling tank, and discharged through the engine water jacket. The engine temperature was regulated by a bleed of relatively cold water from the main service supply. Accurate control of both inlet and outlet temperatures was maintained and checked, by means of thermometers situated in the inlet and outlet lines. The engine itself was mounted on a universal test bed in conjunction with a Heenan and Froude dynamometer type DPX2.

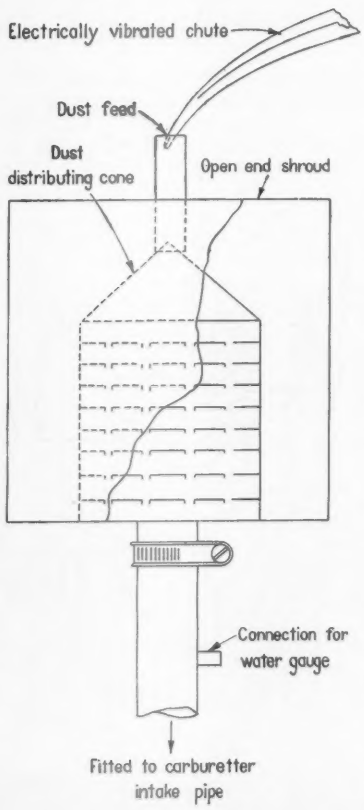


Fig. 1. Dust feed to air filter.

For both trials a standard Vokes Micro-Vee air filter model No. C1 was fitted in conjunction with a dust feed apparatus. The air filter consists of a corrugated felt filtering element supported by wire gauze, and housed in a cylindrical louvred casing. The dust feed was regulated by means of the apparatus shown in Fig. 1.

The oil filter used in the first trial was the Vokes model No. E.30/7915 standard full flow type, which consists of a corrugated filtering element in cylindrical form supported by a wire gauze, and fitted with end caps. This assembly known as the filter insert, is housed in a pressed steel bowl with a cast cover. The unfiltered oil enters the centre of the insert assembly, and flows from inside to outside through the filter element.

In the second trial a partial flow filter of a well-known proprietary make was fitted in accordance with the manufacturer's instructions. It consists of a cylindrical container with a central perforated tube, the space between being filled with highly compressed cotton waste. Unfiltered oil, bled from the pump delivery, enters the filter case, and passes, through the cotton waste element to the centre tube outlet, whence it is returned direct to the sump.

Stocks of new pistons, rings, cylinder liners, bearing shells, valves, valve guides, and crankshafts were obtained respectively from the same production batch, and were selected so that all components used in both trials were as nearly identical as possible. The pistons, rings, and liners were supplied by Wellworthy Ltd., and were checked for composition and hardness. The rings were also checked for spring tension, i.e., the force required to close the gap.

The new Standard engine was dismantled, and fitted with liners which were carefully machined to set dimensions, and inserted in such a manner that they could be removed and renewed prior to each test. This procedure was adopted so that identical cylinder surfaces could be produced for each trial. Before the engine was assembled, surface profile graphs were made from the new surfaces of the centre main bearing shells, piston, and compression rings, the original piston and ring assemblies being replaced by those obtained from Wellworthy Ltd. All parts were given

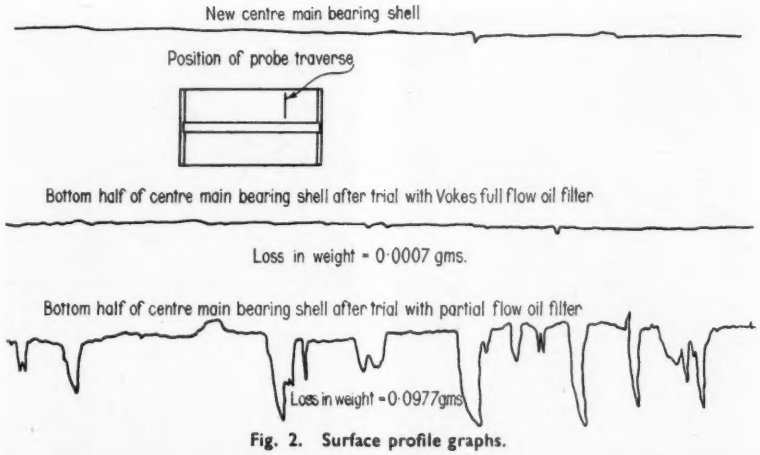

New centre main bearing shell
Position of probe traverse
Bottom half of centre main bearing shell after trial with Vokes full flow oil filter
Loss in weight = 0·0007 gms.
Bottom half of centre main bearing shell after trial with partial flow oil filter
Loss in weight = 0·0977gms

Fig. 2. Surface profile graphs.

identification marks and the following data recorded :

(a) Dimensions of bores and pistons
(b) Weights of all piston rings
(c) Weights of all bearing shells
(d) Dimensions of crankshaft journals
(e) Dimensions of valve stems and weights of valve guides

The engine was checked for valve spring tension, tappet clearances, ignition timing, carburation, etc. These adjustments were recorded, and used as standard settings for each trial.

Twenty gallons of a proprietary non-additive lubricating oil were purchased and examined in the chemical laboratory, as was also a large quantity of fuel, and set aside for the tests. A weighed quantity of the oil was inserted into the engine sump until the level was up to the "full" mark on the dipstick.

The Vokes full flow oil filter was fitted in the circuit between the discharge of the engine oil pump and the crankcase oil gallery so as to filter the whole of the delivery from the pump.

Test schedule

The engine was first run for six hours at one quarter full load, and then six hours at half full load. During this time the dust was fed into the air filter at a rate of 1 gramme per hour. During the whole engine running time, water inlet and outlet temperatures were maintained at 30 to 35 deg C and 78 to 82 deg C respectively. The sump oil temperature was controlled at 70±2 deg C by means of water-cooled coils. The exhaust back pressure was maintained constant for all engine trials.

After these runs, which served to run in the engine, a power curve was taken in order to check the engine's mechanical condition and performance, by measuring its maximum power output, and fuel consumption at several different speeds. The engine was then run for 45 hours at 2,500 r.p.m. and 18 b.h.p., during which time the dust feed was supplied at a rate of 0·5 grammes per hour. Make-up oil was weighed and added at regular intervals to the "full" mark on the sump oil dipstick.

When the test was completed the engine was dismantled. The components (listed (a) to (e) above) were weighed or measured, piston crown, and combustion chamber deposits were weighed and analysed, bearing shells and No. 4 piston were photographed, the used oil was weighed and examined, and finally the used components were sent to the National Physical Laboratory for surface profile graphs to be recorded.

For the second trial the used liners of the engine block were removed, and replaced by new ones from the initial stock. They were then machined to the same set dimensions as for the first trial. The engine was assembled with new pistons and ring assemblies, crankshaft, bearing shells, valves and

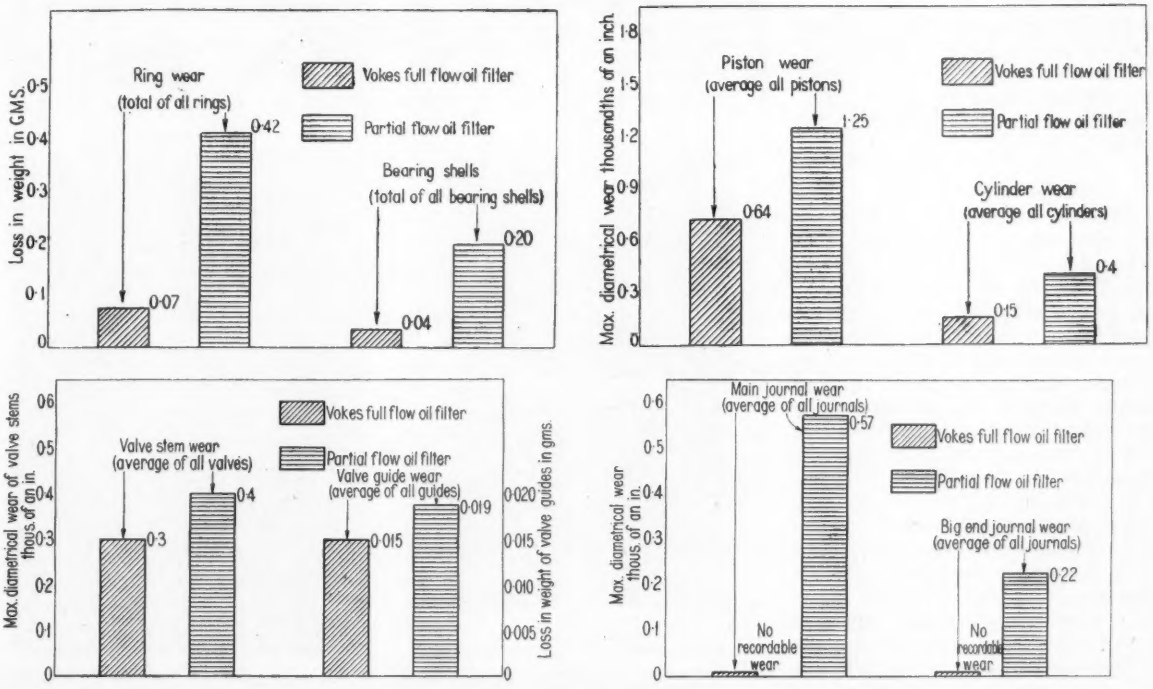
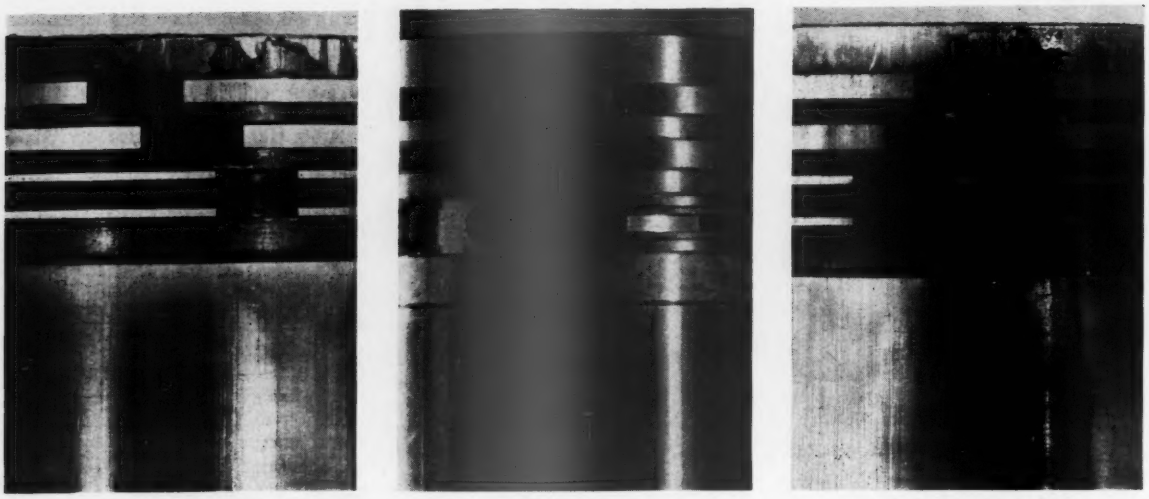


Fig. 3. Component wear charts.



(a) With full flow filter. (b) New. (c) With partial flow filter.

Fig. 4. Pistons and rings.

valve guides. All these components were taken from the initial stock so that they were identical with those employed for the first trial. All the above parts were given identification marks, and measured or weighed as in the first trial. The engine was then equipped with the partial flow type oil filter, and the procedure outlined above for the first trial repeated.

The comparative results as far as wear is concerned are summarized very clearly by the charts shown in Fig. 3. Therefore no further explanation of them is necessary except to mention that wear figures based on loss of weight are for obvious reasons a more accurate assessment than those based on linear measurement. With regard to the general condition of the engine the following points of interest are noted.

Piston condition

The No. 4 piston from each of the two trials, which is representative of all the pistons, is shown in Fig. 4. From this it can be seen that the general appearance and condition of the piston from the test with the full flow filter fitted is the better of the two. Although this piston shows heavier carbon deposits on the lands, the piston ring faces have a better polish, and the machining marks on the skirt show less wear in this region. In Fig. 5 and Fig. 2 it can be seen how much greater is the abrasion of the main bearings when the partial flow filter is used, as compared with that obtained with the full flow filter. This effect is particularly noticeable in the area adjacent to the oilway.

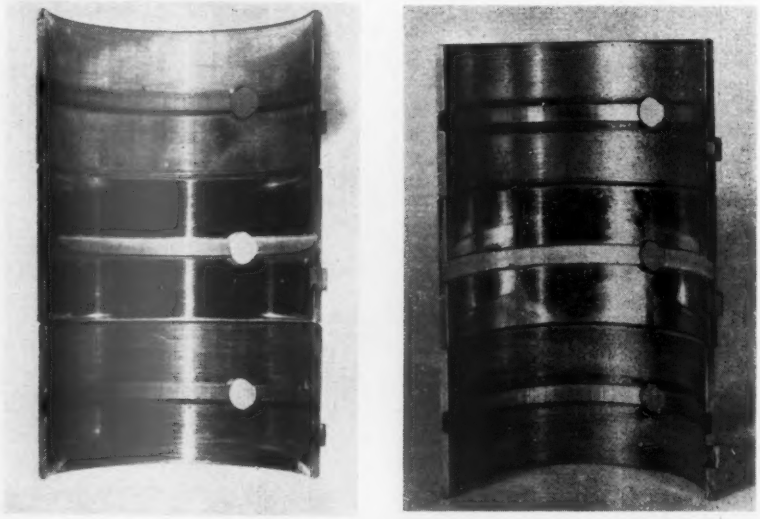
The full flow filter gave a reduction of 7.6 per cent in the weight of the combustion chamber deposits compared with those recorded using the partial flow filter. The composition of the deposits does not show a great variation except in the silica content, which is 2.5 per cent with the full flow filter, and 17.23 per cent when using the partial flow filter.

The photomicrographs (Fig. 6) of specimens of used lubricating oil from the two tests, show that the partial flow filter gives a visibly clearer lubricant, but it can be seen that the oil still contains a large number of particles of approximate size 1 to 10 microns. These are capable of causing abrasive wear, and would account for the higher rate of wear experienced with this filter. On the other hand, the lubricant from the full flow filter test, although less transparent, shows no particles in excess of 1 micron in size.

Conclusions

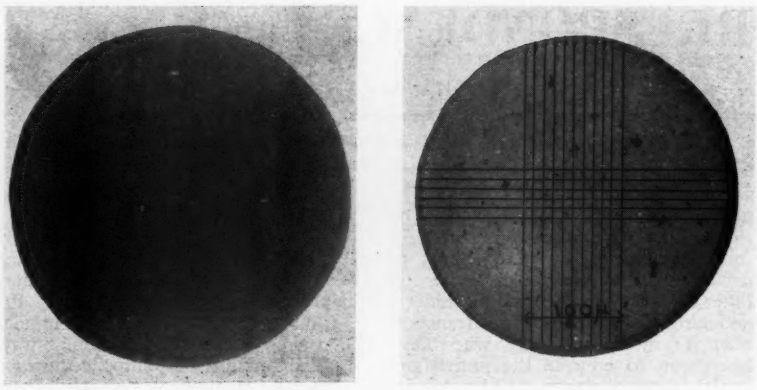
These two tests give fairly conclusive evidence that of the two filters tested the full flow type, by filtering the whole of the oil supply, prevents wear by arresting positively the larger foreign bodies, i.e., those of sufficient size to span the oil film between the moving surfaces. Over a period of time the partial flow filter will remove a large proportion of the solid matter, composed mainly of carbon particles, and maintain the oil in a visibly cleaner condition.

The relative conditions of the pistons in Fig. 4 raises the question whether in a test equivalent to a run



(a) With full flow filter. (b) With partial flow filter.

Fig. 5. Centre main bearings.



(a) With full flow filter. (b) With partial flow filter.

Fig. 6. Photomicrographs of oil.

of about 2,500 miles the carbon deposit with a full flow filter is more serious than the wear. It is suggested that this might be a subject for further investigation. At first sight it would appear that this deposit of carbon is not consistent with the reduction in weight of cylinder deposits when the full flow filter is fitted. This is probably explained by the fact that the reduced cylinder, piston, and ring abrasion gives rise to a smaller passage of oil into the combustion chamber, whereas a carbon deposit is more readily formed on the piston lands, etc., due to the fact that there are more nuclei of carbon in the oil on which the deposit can grow.

It is suggested that a full flow type filter gives a good engine protection, but a partial flow filter, while maintaining the oil in a clean state, gives relatively slight protection from wear.

The conclusions that can be drawn from these tests are limited by the fact that only two filters were tested. The evidence at least indicates that more information is desirable regarding the relative merits of the two systems. Further work on the subject is evidently needed.

QUANTITATIVE-MEASUREMENT OF DETONATION INTENSITY

A METHOD suggested by G. E. V. Jonsson in *IVA Tidskrift for Teknisk - Vetenskaplicforskning*, 1950, Vol. 21, No. 1, and used since 1941 for recording the first differential of gas pressure in the cylinder by means of a piezo-electric crystal, is discussed theoretically. The desirable properties of pick-ups for detonation research are dealt with, and a pick-up of new design, insensitive to mechanical vibrations, is described, with an account of the principles involved in the construction of the associated measuring equipment.

The pick-up is operated directly by the cylinder gases, and its electrical output is proportional to the rate of change of pressure (dp/dt). The detonation meter, however, is unaffected by low-frequency changes in dp/dt occurring in normal combustion, and it responds only to the high frequencies associated with detonation. The meter reading is proportional to the mean value of dp/dt during these high frequency oscillations in the burning gas. It is possible to detect the very weak incipient detonation appearing before audible knock, and to measure quantitatively the detonation intensity as a function of compression ratio. The slope of these intensity curves depends not only on the operating conditions but also on the chemical constitution of the fuel, which accounts for large differences in octane rating obtained for a given fuel by different methods. It is also shown that the octane rating depends on the detonation intensity used as a reference. The intensity curves would seem to provide a means for evaluating the influence of the operating conditions on the tendency to detonation, which in practical running may be appreciably different, even with fuels of the same octane rating. The possibility of measuring objectively the influence of different operating factors on detonation intensity also makes the method described well adapted to combustion chamber research and engine testing. (*M.I.R.A. Abstract No.* 5202.)

BOUNDARY LUBRICATION OF STEEL

BOUNDARY lubrication of steel by blends of acids, esters and soaps in mineral oil has been studied by means of the four ball machine, and reported by W. Davey in *Industrial and Engineering Chemistry*, September, 1950. Naphthenic, stearic, palmitic, oleic, myristic and capric acids, their ethyl and glycerol esters and their soaps with various metals were used. Results are reported as friction-time curves and as the wear-load ratio, which is the number of kilogrammes necessary to produce a wear spot of one millimetre diameter on the fixed test balls.

Blends of acids and esters are better lubricants than straight mineral oil and generally acids are superior to esters (e.g., 1 per cent by weight of myristic acid is equal in wear reduction to 5 per cent by weight of ethyl myristate). This is explained by the fact that acids reduce friction by physical absorption on the metal and reduce wear by formation of soap films of low shear strength. With esters, soap formation can only occur under extreme conditions of load and speed, so that lubrication only takes place through physically adsorbed films of lubricant.

With metal soaps, it was found that those soaps containing a metal less electro-positive than iron (copper, lead) showed a lower coefficient of friction and a higher load-carrying capacity than soaps containing a more electro-positive metal (sodium, calcium). It is suggested that lead and copper soaps react with steel to deposit the metal and form an iron soap. This iron soap forms a tightly adherent film which is a very effective boundary lubricant.

The addition of sulphur markedly improves the lubricating properties of acids, esters and soaps. This is because a sulphide film is formed which facilitates adsorption of the polar compounds. Acid sulphur blends fail at high loads probably because of breakdown of the sulphide film. (*M.I.R.A. Abstract No.* 5132.)

CORRESPONDENCE

Correspondence on subjects of technical interest is invited. The name and address of the writer must be given, though not necessarily for publication. No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter. If a reply by post be desired, a stamped addressed envelope should be enclosed.

DIESEL ENGINES

SIR,—The article in the "*Automobile Engineer*" for February 1951 entitled "Diesel Engines" makes many points of interest both to designers and operators. It is perhaps natural that the writer has given pride of place to electrical and fuel injection equipment, but many of the statements made cannot go unchallenged, and we would appreciate the courtesy of being allowed to refer to a number of them. On the electrical side, the references to the maintenance difficulties experienced with Axial type starters do not tally with the facts of present day service department records and our own Eastern representatives' experiences. The evidence shows that clutch and pinion wear are only a slight amount greater than in this country, and that with reasonable maintenance of the type which is available with trained native labour, there is no more trouble with the Axial starter than there is here. The starter is not a difficult item to service as the writer of the article would have us believe. After all, the engine itself has to be serviced, and there is no work on the starter in any way more difficult than this. In any case, adequate training of the staff must be given in this as in any other subject, and to permit abuse of equipment to take place is merely to show the absence of proper management.

As regards batteries, the troubles may very well be due to the fact that many of the batteries in India and Pakistan are of local manufacture and of poor quality. Licence restrictions and tariffs largely prevent the import of British made batteries. Of course, proper instruction of employees is the only answer to the abuse problem, but the abuse would be just as prevalent with petrol engined vehicles as with Diesels.

Of the alternative methods of starting mentioned, these can scarcely be regarded as available today; the small auxiliary petrol engine as used on large American tractors, etc. (the majority of U.S. Diesels are of the pre-chamber type, and are not inherently good starters) is not applicable to the smaller British automotive type of engine. As regards the hydraulic starter, this has only recently gone into production and has yet to prove itself. It is not in use on transport vehicles, but with development it may well prove a satisfactory alternative to the electric starter.

Coming to fuel injection equipment, the suggestion that some revolution in design is necessary in order to free the fuel injection pump from the mystery with which it is now enshrouded is nonsense in our opinion. It may be right to say that the pump is a mystery to the untrained native operator, or driver, but then, so are a good many other things on engines and machinery, and surely it is the task of the transport manager, more particularly when he is European, to educate the native and explain some of the so called mysteries with which he is beset. This task may well be difficult, but the country is becoming more and more industrialized and the job cannot be shirked. That it can be done, we have proved in our own depots where are employed several very good mechanics working to quite high standards. It is becoming widely appreciated that considerably more service stations will be necessary as the number of Diesel engines in India and Pakistan increases, and that manufacturers must do their utmost in disseminating up-to-date service information to transport managers, engineers and operators throughout these and similar countries.

Dealing with the author's four points suggested in order to increase the life of fuel pumps between repair periods, the following may be said :—

(*a*) The calibration system suggested comprising a home-made bench with a system of "squish" arranged behind a graph-like screen is, in our opinion, so useless as to be unworthy of further comment. In any case, the "Squish" method can only be applied to pintle type nozzles, if indeed it can be applied at all, and variations in fuel delivery between elements of up to 50% will get through such a test. The use of such a system would be strongly opposed by us. If a home-made test bench can be made to run at speed, such as is suggested, then even a native could surely be trusted to collect fuel issuing from injectors into a suitable receptacle with a far more accurate result in calibration than could possibly be obtained with the system described by the author.

(*b*) Phasing is not a process which is required to be carried out often, and we have found that the majority of mechanics seem quite capable of following our procedure. If, as it is to be supposed, the author's native operator is capable of adjusting the tappets on his engine with reasonable success, then surely it is not too much to expect the same man to master the adjustment of fuel pump tappets after some very elementary training.

The drawing on page 61 showing cracks in the pump housing caused by "ham-fisted" handling during phasing is, in our opinion, incorrect, as such cracks do not occur. Cracks in pump casings may be caused by "ham-fistedness" in the tightening of delivery valve holders, but they are higher in the casting than the illustration shows; any equipment can be damaged in a similar way. The phasing of pumps requires a certain amount of training as we have suggested, and it is up to the management concerned to handle this problem in the same way as any other maintenance operation.

(*c*) On filtration, much of what the author says is true and good common sense. The problem of getting clean fuel oil to the point where it enters the injection pump is one to which we have devoted very considerable attention, as it is one of the major points in successful Diesel operation. We ourselves are doing all possible by means of leaflets and wall charts to explain the correct method of servicing filters. While it is no doubt true to say that in many cases filter cleaning has led to damage of pump elements due to careless handling, permitting dirt to enter the clean side of the filter, we cannot agree that the periods of uninterrupted running without filter servicing of one year for cloth element filters and two years for felt element filters, are reasonable. The filters would become completely clogged at very much shorter periods in dusty regions, and the recommendations do not tally with the short mileages the author mentions of 15,000 to 50,000 obtained with fuel injection pumps.

As regards fuel cleanliness, the author's remarks on matters affecting distribution and storage are very pointed. Undoubtedly, the question of suitable storage tanks for fuel oil is one which demands greatest attention. Where oil is delivered in a perfectly clean condition it can still be badly contaminated in unsuitable tanks. On the other hand well-designed storage tanks will do much towards preventing wear by allowing sedimentation to take place. Even with clean fuel in the tank, dust can still enter through air vents and the need for adequate filters immediately preceding the injection pump is evident.

(*d*) The author seems to have a very jaundiced view of the use and durability of the pneumatic governor. The troubles reported with many hundreds of governors of this type, now in service in the East, are to the best of our knowledge in no way excessive, and the experience of some of our largest

engine manufacturers shows that the incidence of failures with the pneumatic governor diaphragms is extremely small, and certainly not sufficient to justify any changes being made.

The author seems to display a lack of appreciation of the subject of governing in his references to weights having given place to pneumatic control, and now the latter being under threat of displacement by hydraulic control. No reference is made in speaking of engines which might run away due to damaged vacuum pipe or diaphragm, to the use of the overriding hand stopping control.

The system of dustproofing now being applied to fuel injection pumps for use overseas, especially in dusty conditions, should go some way totowards overcoming some of the troubles experienced.

We would emphasize that tampering with injection pumps is a thing which should not be tolerated anywhere, let alone overseas, and that adequate instruction and supervision must be given. There is no especial difficulty in training operators to handle fuel injection equipment, and there are many parts of the modern motor vehicle where a similar degree of knowledge and skill are required for proper maintenance. Work on engines themselves, such as re-boring, etc., calls for care and a standard of workmanship of a relatively high order, and if this can be tackled satisfactorily there is no reason why fuel injection equipment maintenance work should not be done with equal success.

C.A.V. LTD.,
Acton, London, W.3.

POLAR MOMENT OF INERTIA

SIR,—Except behind the Iron Curtain, evidence is based on facts, not on opinions; such facts in scientific evidence being numerical and obtained by experiment.

No such evidence is forthcoming in your comment on my letter and the discussion must remain limited to one of opinion and theory.

Theory is more convincing if backed by reasoned argument and, if possible, by calculation. Again this is not to be found in the quotations from the B.I.O.S. Final Report No. 1755.

On the contrary, the views of the Mercedes technicians, as reported, are contradictory and the compiler's grasp of first principles appears to be shaky.

So much so that it is due to the memory of the late Dr. Porsche to point out that this distinguished engineer would not have thought that rotational inertia forces are proportional to the distance of a mass from the centre of rotation when in fact, they are proportional to the square of that distance.

I am not attempting to dispute the contention that the steering of the Auto-Unions cars may have lacked perfection.

This may have been due to one or more different causes but I still await evidence to prove that it was due to the low moment of inertia of the cars and that their gifted designer was not right in adopting it.

G. SARTORIS,
A.B.C. Motors Ltd.

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

A Century of Technology

By specialist authors, under the Editorship of Percy Dunsheath, C.B.E.

LONDON: HUTCHINSON'S SCIENTIFIC AND TECHNICAL PUBLICATIONS, Stratford Place, W.1. 1951. 5½ × 8½. 350 pp. Price 15s.

As the title implies, this book sets out to cover the development of the sciences in their applications to industry during the past 100 years. In its own way, it is a highly educative work and a mass of scientific history is embodied in tabloid form. Now that automobile engineers have contacts with so many trades and sciences, a brief comprehensive volume such as this is of definite interest. Being a history, it deals with the origins and the basic discoveries from which the bulk of industrial development has sprung. New materials and methods are better appreciated and understood against their historical background. Each of the many subjects is dealt with by a specialist, and typical chapter headings are as follows:—

Iron and steel
Non-ferrous metals
Non-metallic materials, including plastics and rubber
Chemicals
Textiles
Production engineering
Fuel and power
Transport, etc.

Taking an example within the purview of the engineer, the section on production engineering provides a suitable case. Here, the customary plan is followed and the early days are first briefly surveyed. The work of the pioneer machine tool builders such as Maudslay, Naysmith and Whitworth is referred to, with some account of the early tools they so brilliantly evolved. After these machines had developed more fully over the years, the next step was the coming of the integral electric drive and the wide employment of hydraulic controls.

Cutting tool development marks the next considerable step forward, following which come the numerous auxiliary metal forming and fabricating processes, including the welding techniques, powder metallurgy, presswork, etc. Really high precision measurement, full standardisation and elaborate methods of mechanical handling, together with all modern developments are fully covered. In short, as with other sections in the book, a very complete though highly compact picture is provided.

Despite the necessarily brief character of the various surveys, however, it has been possible to include a surprising amount of detail. The book is well illustrated and the various subjects are covered in sufficient detail to appeal to the engineer. Although doubtless directed primarily to the enquiring layman, there is much that would interest those with a more particular interest in the various subjects. In works of this description the matter of references is a vital factor in exploiting the utility of the volume, and it is satisfactory to note that a very full index is provided consisting of 15 pp. of two column small type.

Protective Coatings for Metals

By J. W. Gailer, M.Sc., F.R.I.C., A.I.M., and E. J. Vaughan, M.Sc., F.R.I.C.

London: CHARLES GRIFFIN & Co. LTD., 42, Drury Lane, W.C.2. 1950. 6¼ × 9. 261 pp. Price 24s.

The value of protective coatings for metals has been appreciated for decades and many industries have been intensely interested in the subject. It was during the last war, however, and immediately afterwards that the matter really came into prominence and several books have appeared on the subject. In war time, equipment had to be shipped to foreign countries where it had to withstand every type of atmospheric condition, including water. Since then, increased exports all over the world have emphasised the importance of protective coatings. In the motor industry, it is essential that parts not painted, or parts liable to lose their protective enamel coatings, should not rust or deteriorate. The subject is perhaps more of interest to the motor industry than to any other, although all are affected to some degree.

There are many types of protective coatings for metals and many methods of applying them. Manufacturers must be competent to decide the best and the most economical ones for their specific purposes, and they must also have the means of testing them after they have been applied. This book gives details of the most important protective coatings and also of testing methods.

Included in the book is a summary of all these methods, in the form of valuable tables. They give the physical and chemical properties of the coatings mentioned in the text, their resistance to high temperatures, a general guide to the principles involved, brief details of methods of inspection, and other relevant points. The rest of the book is an amplification of this information.

The first section on Metallic Coatings covers the principles of electrodeposition, cadmium, zinc, tin, nickel, chromium, lead, copper, speculum, etc., coatings being dealt with separately. In each case suitable bath mixtures, working conditions and methods of inspection are given. Dry Processes include hot dipping and cementation, covering hot tinning, galvanizing, lead alloy and aluminium coatings, sheradising, chromising and calorising. A few notes are included on metal spraying.

The second section is devoted to Non-

Metallic and Duplex Coatings. Here will be found details of Parkerising and Bonderising, with notes on Coslettising and Walterising. Processes for treatment of aluminium, magnesium, zinc, and their alloys are all briefly dealt with. The last two chapters are concerned with paints, lacquers, enamels, and temporary protectives.

The appendices consist of notes on general aspects of methods of inspection including salt spray, immersion, mechanical, electrochemical, weathering, and optical tests. Some illustrations of the equipment employed are included.

Manufacturers and inspection departments will find in this book sufficient information to help them to decide upon the most suitable coatings for their specific purpose. Others, interested in the subject, will find the list of over 200 references to published papers on the subject invaluable. The book is brief in many respects since obviously a volume could be written on each section, but the facts given are clear and concise and no space is wasted.

The Story of Vauxhall 1857 to 1946

By L. C. Darbyshire.

Luton: VAUXHALL MOTORS LTD. 1950. 7 × 9. 51 pp.

Vauxhall Motors Ltd. have published a book describing the growth of their firm from the early days as marine engine manufacturers to the present-day producers of the modern car and commercial vehicle. Developments from the very earliest model are traced, so that the story is more than a history of one firm, for it gives a brief outline of the progress of motoring in general. The first model put on the market by Vauxhall was a single cylinder car, weighing just over 5 cwt. and "steered by a comfortably placed tiller actuated by the left hand". There were two forward gears, but no reverse. This was in 1903, and an attempt was made a little later to adapt a hansom cab body to a Vauxhall chassis, with the steering controls in the place where the cab driver normally sat. This project met with little success and was soon abandoned. The 5 h.p. car was followed by one of 6 h.p. and so the story continues, with details of improvements, experiments, trials, and the successes won in them, through two wars to the present time.

It is amusing to learn of the number of spare parts the early drivers were recommended to carry with them, which almost amounted to a complete car, but not the least interesting part of the book are the many photographs and a chart at the end provides information, including prices, of all the standard ranges produced from 1903 onwards.

The Practical Engineer Pocket Book, 1951

Edited by N. P. W. Moore, B.Sc., A.C.G.I., D.I.C., A.M.Inst.F.

London: SIR ISAAC PITMAN & SONS, LTD., Kingsway, W.C.2. 1951. 5½ × 3¼. 744 pp. Price 8s. 6d.

The record of sixty three years of publication is one that proves the value as well as the popularity of any pocket book. This year, however, the publishers and the editor of this pocket book have changed and as a result, many alterations have been made, although the general layout and material are retained. The type has been completely reset and is easier to read. Many illustrations have been redrawn and some of them are more up-to-date. The sections have also been re-arranged in a more logical sequence. The section on internal combustion engines now includes information on the properties of compression ignition engine fuels, and notes on supercharging have also been added.

An entirely new section on gas turbines is also included and this has notes on typical simple open cycles, open cycles with heat exchangers, and simple closed cycles. Notes are given about materials and fuels employed for gas turbine construction and running respectively. Other new material concerns basic data on the proposed unified standard screw threads and this section will be extended in future editions when the specifications are finalized. A further addition this year is a list of German translations of technical words and terms, augmenting the French and Spanish ones.

Contents that reappear, or have been revised include those dealing with such subjects as friction and power transmission, pyrometry, metallurgy, steam generation, locomotive practice, gas and oil engines, air compressors, hydraulics, modern lubrication, machine tools, etc. There is much information not usually included in the general purpose pocket book, as for example, frequent mention of lubricating details, this being a legacy of the previous editor, Dr. Stuart.

Ripper's Heat Engines

Revised by A. T. J. Kersey, A.R.C.Sc., Wh.Ex., M.I.Mech.E., F.Inst.Fuel.

London: LONGMANS, GREEN & CO. LTD., 6 and 7, Clifford Street, W.1. 1950. 4¾ × 7¼. 363 pp. Price 7s. 6d.

First published in 1909, this third edition, after numerous impressions, has brought the subject matter right up to date. An attempt has been made to indicate some of the more recent applications of scientific principles and experiences with new methods of construction.

The first chapters are an introduction to heat, work, heating, cooling, formation of steam, expansive working, etc., and obviously include such subjects as latent heat, properties of saturated steam, etc. The reciprocating steam engine is dealt with, and separate chapters are devoted to crankshafts, drop valve gear, slide valves, condensers and governors, in all cases giving the theoretical principles in a manner suited to the young student who has only elementary mathematical knowledge.

The chapters on fuels, combustion, the furnace, boilers, etc., are now brought up to date and space is devoted to the steam turbine. A chapter is devoted to internal combustion engines. The examples for the student to work out are as useful as ever, and although the book has now been modernized, it is still Ripper's Heat Engines, and it will certainly go on for many years providing students with the elementary information that they require during their early work on this subject.

Diesel Engine Manual

Edited by E. Molloy.

London: GEORGE NEWNES, LTD., Tower House, Southampton St., W.C.2. 1950. 6 × 9. 212 pp. Price 15s.

The publishers state that this book is intended for erectors, installation and plant engineers, and all interested in the practical aspect of diesel engine operation. This is quite true, but to the practical man engaged in one branch of diesel engine work, only certain sections of the book will be of service. For example, some forty pages are devoted to the installation of stationary oil engines, and it is doubtful if those who need to have details of this work will also need to know about automobile diesel engines, except as a matter of general interest. There is also a chapter on the marine heavy-oil engine.

The subject is really too wide to be covered in one book, and the publisher's statement on the dust cover that this is "the most comprehensive, informative and up-to-date work available" can hardly be justified.

Students and young maintenance engineers requiring a general knowledge of maintenance work in connection with all types of diesel engines will find the book interesting. The basic principles and details of fuel injection systems are sound and are well illustrated. In the automobile diesel engines section, notes are given on the maintenance of Gardner and Perkins engines, and it is likely that this section will be most valuable to those who have these engines in their care. Of interest to all diesel engine maintenance engineers are the chapters dealing, respectively, with tests on diesel-type engines and exhaust gas analysis. The tests include descriptions of the use of the Dobbie-McInnes indicators, the Farnboro Indicator, the Froude dynamometer for producing indicator diagrams, measurement of M.I.P., I.H.P., and B.H.P., etc. The last chapter deals with use of the Orsat apparatus, and measurement of implications of exhaust temperatures.

Automobile Workshop Practice

By Staton Abbey.

London: SIR ISAAC PITMAN & SONS LTD. Parker Street, Kingsway, W.C.2. 1950. 4¾ × 7¼. 169 pp. Price 10s. 6d.

Of the many books available to the practical automobile engineer, it might be thought difficult to strike a new note, but this little book cannot fail to be found useful to every maintenance man or motor car enthusiast who wishes to know more about the tools he uses, or which he should use. Commencing with some notes on materials and their properties, the practical man should appreciate the importance of the work of the metallurgist and should realise that without him there would be no motor car to repair. The chapter on reading and using machine drawings includes important information on fits, clearances and tolerances with charts of Newall limits.

The reader is then introduced to precision instruments such as Johansson gauge blocks, micrometers, feelers, dial gauges, etc., and is given sound advice upon the use and care of hand tools. Grinding of drills, angles, speeds and lubricants are all included in basic principles, but sufficiently to gain the interest of any worthwhile mechanic in his job. Amongst the larger machines described are lathes, boring bars and grinding machines, and, of course, cylinder reconditioning equipment including boring bars and honing machines are included and illustrated.

Valuable information is given on renewing bearings and bushes although whether sufficient detail can be given on remetalling bearings in the short space available is doubtful. The author probably does not intend this, and the notes cannot fail but be helpful to those who may have to undertake this work. Under Screw Threads and Locking Devices will be found notes on screw cutting in the lathe, but here again, probably the information is too brief to be practical. The rest of the book is devoted to soldering, brazing, welding, riveting and the heat treatment of metals, and a number of useful tables, facts and figures are included.

CURRENT PATENTS

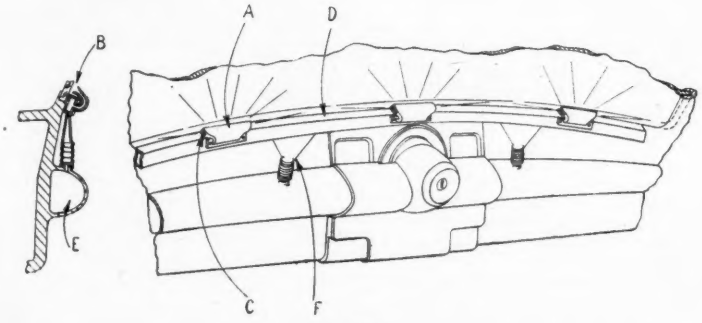
A Comprehensive Review of Recent Automobile Specifications

De-Misting Windscreens

IN this method of de-misting a windscreen, a plurality of elongated nozzles with additional lateral orifices is used to distribute heated air from a single supply duct. In the example, three such nozzles A are spaced along the lower edge of the windscreen inside the vehicle. Each nozzle has a mouth B of narrow rectangular section arranged parallel to, but slightly spaced from, the plane of the screen. The narrow side walls of the nozzles are cut away at the mouth, as at C, to allow subsidiary streams of air to be projected along the lower edge of the screen. Extending along the length of the screen is a rectangular supply conduit D, which may also serve as the lower rail of the windscreen frame.

The air supply may be obtained from any convenient source and may be induced or supplied by a mechanical blower and the temperature raised in a heat exchanger or by electric equipment. It is delivered to a duct E formed in the facia board from which two connections F establish communication with conduit D.

It is understood that this equipment may be used to supply cool air to the interior of the vehicle when it is unnecessary to deliver warmed air for de-misting. *Patent No.* 637203. *Humber Ltd. and L. W. Mell.*



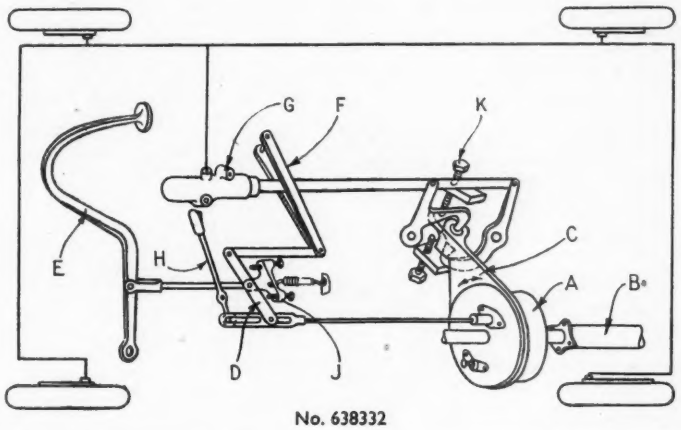
No. 637203

Servo-Assisted Hydraulic Brakes

IN this system a mechanically-operated transmission brake functions as a servomotor to apply pressure to the master cylinder of a hydraulic braking system. The transmission brake A, mounted on the output shaft of the gearbox or on propeller shaft B, has a pair of shoes of the ordinary non-servo type mounted on a back plate carrying a radially projecting torque arm C. The expander of the shoes is actuated by a tension rod coupled through a pin-and-slot connection to one end of a balance lever D. Centrally this lever is linked to brake pedal E, while its opposite end is linked to a forked lever F, pivoted on a fixed point and connected to the push-rod of master cylinder G.

The free end of torque arm C operates between two opposed bell-crank levers, pivoted about fixed points and connected to the master cylinder push-rod, so that movement of the torque arm in either direction applies pressure to the fluid in the master cylinder.

The transmission brake can be used for parking by means of hand lever H, connected through a lost-motion device. Adjustable stops on a crosshead J limit the angular movement of balance lever D about its pivot, so that the transmission brake can still be applied by the pedal in the event of a hydraulic failure and, conversely, the hydraulic brakes can be operated in the event of a failure of the transmission brake mechanism. Movement of the torque arm is also limited by adjustable stops K, so that the transmission brake can be operated effectively by means of the hand lever. *Patent No.* 638332. *Girling Limited.*



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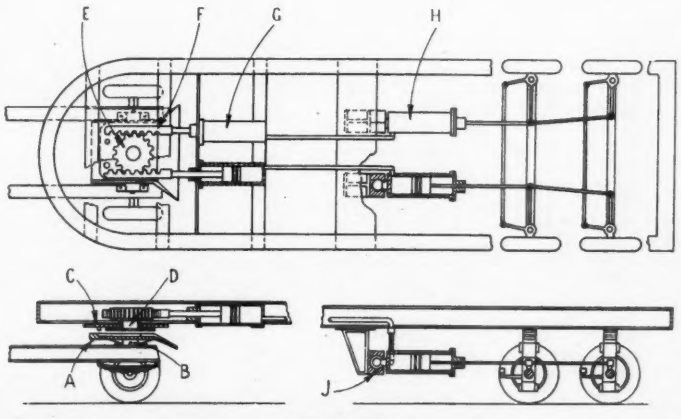
Steering Articulated Vehicles

THE trailing wheels of this articulated vehicle are dirigibly mounted and controlled by a hydrostatic mechanism operated from a gear on the turntable pivot. At the rear of the tractor a support member A, tiltable about a transverse pivot B, receives the turntable ring C on the front end of the trailer frame. In a bearing concentric with ring C is a vertical shaft D, flanged and bolted (detached in the illustration) to member A and carrying a keyed gear E. Racks F engaging this gear move pistons in pressure cylinders G mounted on the trailer frame.

These cylinders are in communication with actuation cylinders H, suspended by ball supports J from frame brackets, and their pistons are linked to the steering arms of the trailing axle or axles. *Patent No.* 636135, *N.V. "Hulsemo" (Holland).*

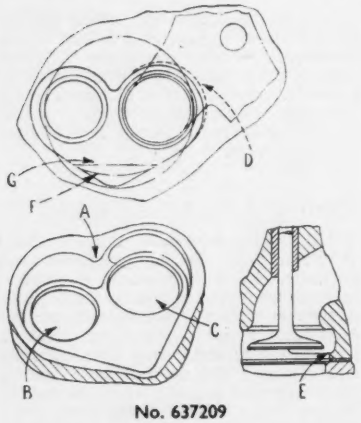


No. 636135

Overhead-Valve Combustion Chamber

THIS cylinder head, particularly intended for vertically-mounted overhead valves is furnished with a lip extending partly around the inlet valve seating to direct the incoming gas towards the cylinder axis. The general contour of the combustion chamber is heart-shaped to provide a projection A which extends into the space between exhaust seat B and inlet seat C. Part of the peripheral wall encircling the inlet seat is undercut as indicated by broken line at D. The wall of the undercut may be concave and leave a lip E extending around approximately 180 degrees of the seat. Usually the wall of the combustion space around the exhaust seat will be parallel to the axis of the cylinder, although in some instances it may locally be concavely curved to offer less restriction to flow. The remaining part of the peripheral wall will vertically be straight or concave, and in transverse section is V-shaped. As required, however, the section may be modified to the curved or flat profiles indicated respectively at F and G.

A precise location for the sparking plug



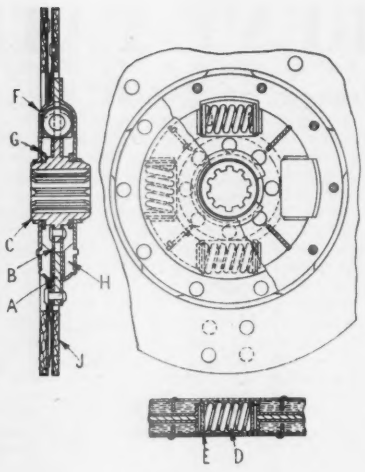
No. 637209

is not specified. It will be placed to be swept by the incoming mixture and may be furnished with a socketed mounting, in order to collect mixture in the neighbourhood of the plug points. *Patent No.* 637209. *Henry Weslake.*

Fluid-Damped Spring Clutch Plate

IN a spring-cushioned clutch plate some means of damping spring action is desirable to smoothen the drive but frictional devices to achieve this end are liable to lose their effectiveness after a period of wear. Furthermore, under certain conditions, the springs are apt to emit sound. Fluid damping by means of heavy oil or grease is proposed to overcome both problems.

The main components are of familiar type, with the outer ring A located between a pair of plates B riveted to the flange of splined hub C. Helical springs D are fitted at both ends with fibre buttons E carrying wear-resistant metal washers to engage seatings formed by outwardly-bent lugs on plates B. Cemented and riveted to ring A is a pair of dished plates F having a close bearing on the hub and sealed by packing rings G to form the annular fluid chamber. This chamber is partitioned by the spring seatings on the inner element and paddle vanes H on the



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outer element, leaving only restricted clearances through which the fluid is displaced in operation.

The illustration shows a twin frictional ring J, spaced by so-called "mush" springs but this is not an essential feature of the design. *Patent No.* 637257. *Dana Corporation (U.S.A.).*

Hydraulic Actuation of Drop-Heads

A FEATURE common to the three arrangements suggested is a hydraulic cylinder fixedly mounted on the vehicle body and operating the movable linkage from a guided cross-head.

In the first, cylinder A is carried on a fitting B incorporating parallel guides for the cross-head C. The main stay D of the hood is pivotally attached to the fitting and connected to the cross-head by a link E.

In the modification, the main stay F is separately mounted on the vehicle body and the cylinder and cross-head guide is so located that when the head is closed link G occupies a position substantially at right angles to the main stay and plunger and thereby serves to lock the stay in its upright position independently of fluid pressure.

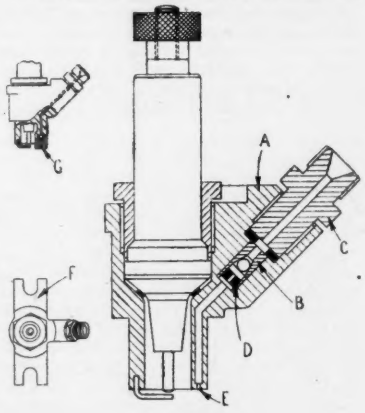
In the third method, link H is not connected directly to the main stay J but to an intermediate point of one of a pair of toggle links K connected to the support fitting and the stay. *Patent No.* 637043, *Wilmot-Breeden Ltd., and C. H. Stephenson.*

Injector for Spark-Ignited Engines

INTENDED for spark-ignited engines running on volatile fuel, this atomizer is suitable for adapting conventional power units to operation with liquid injection to the cylinders. The fuel is supplied in accurately-metered charges in timed relation to the piston movement.

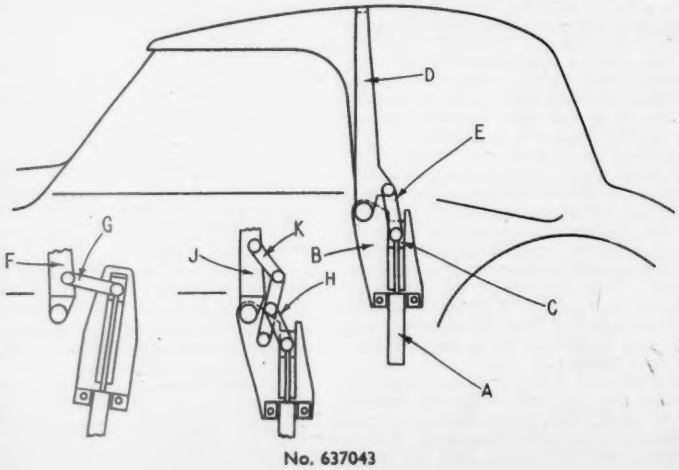
In general the construction follows that of the ordinary spark plug having a detachable central electrode retained by a gland nut. The body, however, is formed with a lateral protuberance A, bored to accommodate a coned seating B for a ball valve and a connector C for the attachment of the pipe line from the fuel pump. An apertured stop D limits the travel of the ball which is shown as seated solely by cylinder pressure but may be supported by a light helical spring. The fuel is delivered by way of a drilling in the wall of the body spigot, constricted at the face of the spigot to form a spray orifice E.

The spigot may be threaded and screwed in the usual spark hole in the cylinder but



No. 637584

preferably lugs F are provided, enabling the device to be secured by two studs. If a larger diameter spigot can be accommodated the spray orifice may be arranged in a demountable jet G or, alternatively, the body may be internally threaded to receive a standard spark plug.—*Patent No.* 637584. *Mono-Cam Ltd.*



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